THE RELATION OF CYLINDER AND BOILER POWER TO LOCOMOTIVE RATING.

Paper read by E. M. Gass, Member, Horwich, on Friday, 4th April, 1919, at Manchester, and on Tuesday, 27th May, 1919, at Leeds.

PAPER NO. 73.

The hauling capabilities of the steam locomotive are of consequence to the Chief Mechanical Engineer, the Traffic Official, the Running Superintendent, the Designer, and all responsible for the operating of train loads.

The Paper is divided under two headings :—

(1) ENGINE POWER, *i.e.*, the power the locomotive is capable of exerting behind the drawbar, and the loads it can haul at various speeds on a straight track and on various grades and curves.

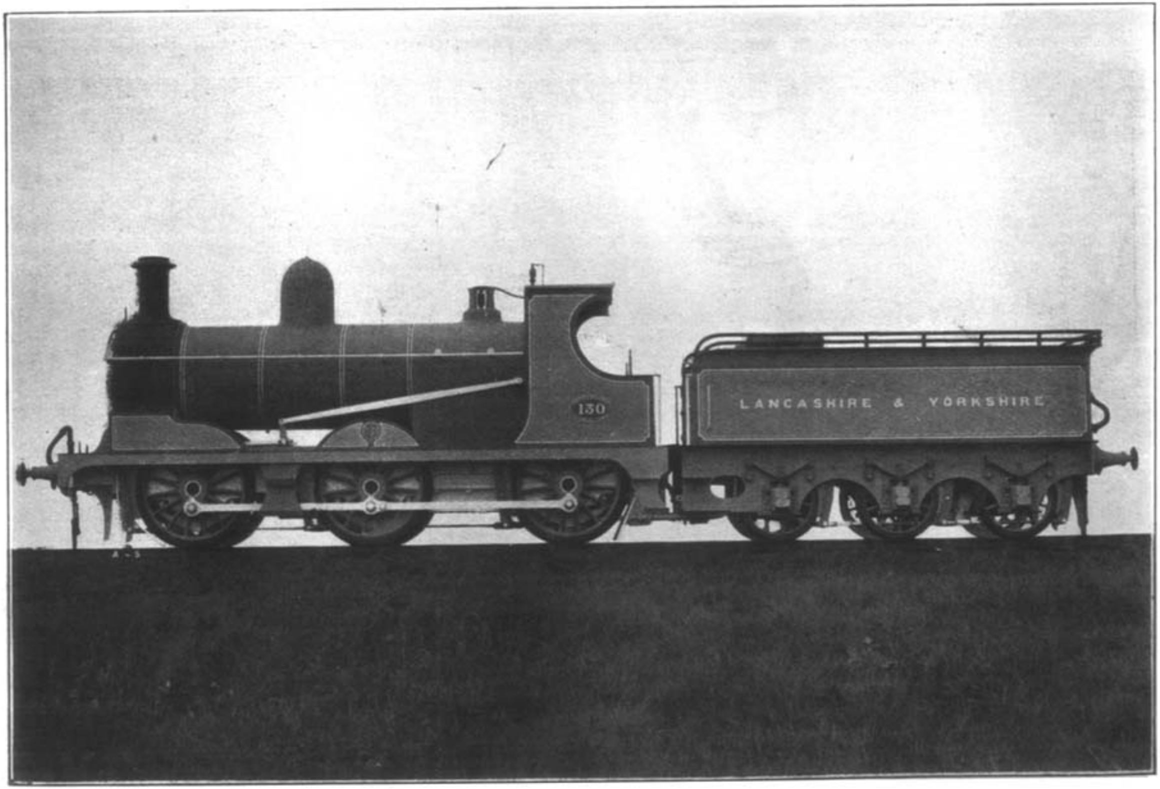
(2) BOILER POWER, *i.e.*, the capacity of the boiler to meet the demands upon it, so that the engine can economically haul its loads at given speeds on various grades.

The deductions are made on experiments carried out on the Lancashire and Yorkshire Railway with goods trains composed of wagon stock of 10½ tons gross weight each (wagon plus load), hauled by locomotives using saturated and not superheated steam.

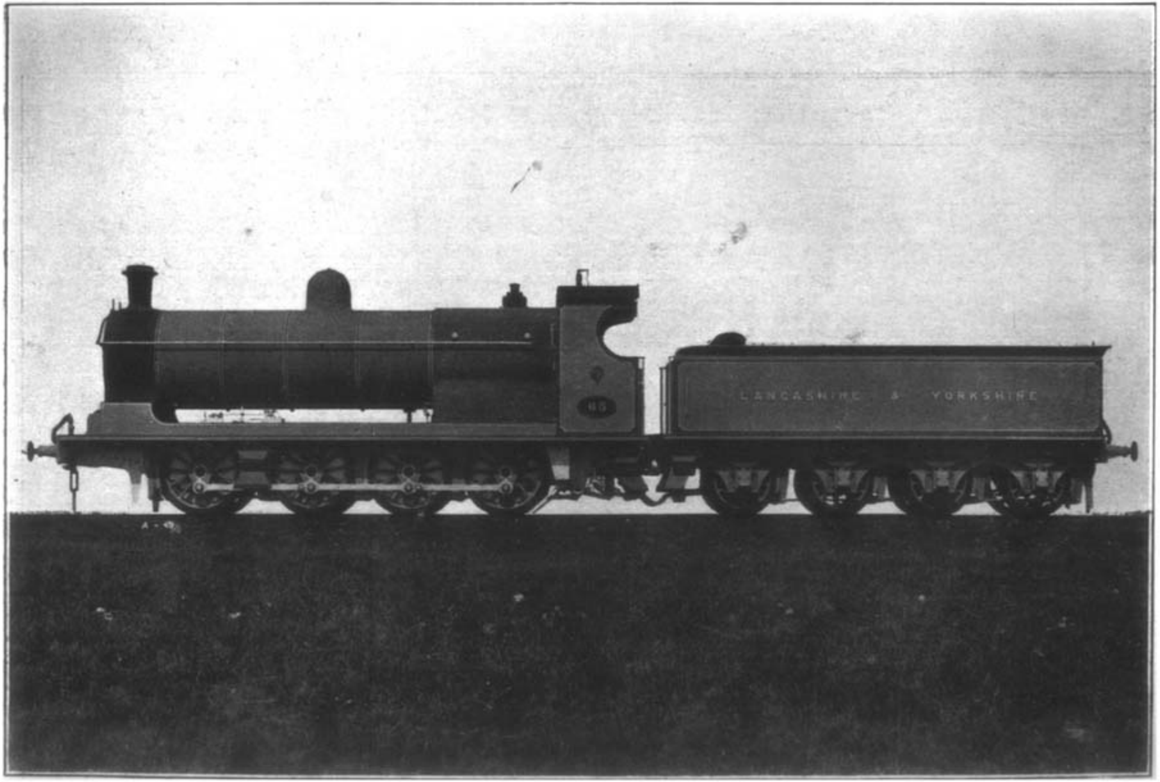
The following are the leading particulars of the two engines employed on the trials (see photographs):—

Class.	0-8-0 Goods.	0-6-0 Goods.
Cylinders ...	20" dia. x 26" stroke.	18" dia. x 26" stroke.
Wheels ...	4' 6" dia.	5' 1" dia.
Working pressure lbs. per sq. in.	180	180

	T.	C.	Q.	T.	C.	Q.
Weight in working order—						
Engine	53	15	3	42	3	0
Tender	42	0	0	26	2	2
Total	95	15	3	68	5	2



" Locomotive Rating."
L. and Y. Ry. o-6-o type.



" Locomotive Rating."
L. and Y. Ry. o-8-o type.

Adhesion at 20 per cent. of weight on drivers	24,097lbs.	18,882lbs.
Ratio of cylinder volume to total heating surface cubic in. to sq. ft.	4.06 to 1	5.4 to 1

ENGINE POWER.

STARTING EFFORT.—There is no fixed rule as regards the pressure factor in the well-known formula :—

Deduced from first principles :—

Work done at tread of wheel = work done by steam on piston.

$$2\pi r T = 2 \times \left(D^2 \times \frac{\pi}{4} \times P \times 2S\right)$$

$$T = 2 \times \frac{D^2 \times \pi \times P \times 2S}{4 \times 2\pi r}$$

$$T = \frac{D^2 \times P \times S}{2r} \text{ or } \frac{D^2 \times P \times S}{W} \text{ (for 2-cylinder simple engines).}$$

T = Tractive effort.
D = Diameter of cylinder in inches.
S = Stroke of piston in inches.
P = Mean effective pressure, lbs. per sq. in.
W = Diameter of driving wheel in inches.

Recently the relation that *P* bears to boiler pressure has been debated upon by leading locomotive engineers, but the Author has no knowledge of any definite figure. Some authorities give *P* as low as 75 per cent., others as high as 85 per cent., of the boiler pressure; the Author has investigated a number of indicator diagrams, and finds 82 per cent. to be a safe figure for calculating the maximum available tractive effort. A recognised standard for P ought to be brought into force. Based on 82 per cent. of boiler pressure, the maximum tractive effort of the two engines under consideration is 28,426 and 20,383lbs. respectively.

CYLINDER MEAN EFFECTIVE PRESSURE.—The factors which contribute to decrease in mean effective pressure as speed increases are :—

(*a*) Reduction in the period of opening of steam port.
(*b*) Reduction in the period of opening of exhaust port; this duration of opening is greater with long than short valve travel.
(*c*) Enhanced compression.
(*d*) Increased steam flow; a particular influencing factor.

Fig. 1 shows three mean effective pressure curves. The *L* and *Y* curve has been plotted from a large number of indicator diagrams taken on the 0-8-0 engine.

The cross sectional areas of the pipes and passages of this engine are :—

Regulator ports	23	sq. in.
Main steam pipe in boiler	28.27	,,
Steam pipe in smokebox	19.635	,,
Steam port	25.022	,,
Exhaust port	63.063	,,
Ratio of steam port to cylinder ...	1 to 12.55	

Referring to the curves, at starting the *L* and *Y* pressure is higher than the American, but lower than Cole's. Cole's curve, however, begins to fall immediately after starting, whereas the American pressure is maintained up to a piston speed of 220 feet per minute. The *L* and *Y* tests proved that the maximum pressure could only be maintained up to a speed of 130ft. per minute; between the speeds of 300 and 900ft. per minute, the *L* and *Y* curve practically falls between the American and Cole's, after which the three curves merge into one.

RESISTANCE.—Engine, tender and train resistance, apart from grade, curve, and wind resistance, comprises two components :—

(1) Engine resistance (internal resistance) of the machinery plus resistance as a vehicle.

(2) Train resistance (vehicles comprising the train), the tender being regarded as a vehicle.

ENGINE RESISTANCE.—Engine or internal resistance is the sum of the frictional resistance of :—

Pistons and piston rods.
Crosshead slides.
Connecting and coupling rod bearings.
Driving axlebox bearings.
Valves and valve rods.
Valve motions.

Considered by some authorities, internal resistance is constant for all speeds, by others, that indicated tractive power influences it. Probably it is somewhere between the two assumptions, and increases with the number of wheels. Starting resistance is much higher than when running at low speeds. The efficiency of the two engines mentioned above was found to be 95.0 and 96.5 per cent. respectively, giving an internal resistance of 23lbs. per ton for the 0-8-0 engine, and $16\frac{1}{2}$lbs. per ton for the 0-6-0 engine, at starting. The internal resistance curves for various speeds are shown by Fig. 2.

TRAIN RESISTANCE.—The components of train resistance are :—Journal resistance, rolling resistance, wind resistance, and miscellaneous resistance (resistance due to oscillation and wheel flange friction).

The largest percentage of wagon train resistance is due to journal friction; this resistance, however, varies with the character and application of the lubricant, the condition and composition of the metals in contact, and the bearing pressure per unit of area. Co-efficient of friction is more than doubled by the use of mineral grease lubricant, when compared with sperm oil, and the oil bath is a much superior method of lubrication than the syphon and pad. The claims of anti-friction metals to reduce bearing friction are numerous. The unit of pressure is of great importance, for if this is excessive the lubricant is unable to find its way between the contact surfaces, and imperfect lubrication is the result. A very useful and valuable paper might be compiled on lubrication of locomotive journals.

Rolling resistance is small, also air resistance at low speeds, but at high speeds it is of great importance. To estimate for wind resistance, as distinct from air resistance, is almost impracticable, nor can miscellaneous resistance be determined. Many experiments have been conducted, and theories put forward, in order to discover a satisfactory solution of the problem, but no two investigators are agreed. Experiments seem to prove that head air resistance increases with the square of the velocity, but the exact constant depends on the area, shape, and general design of the front end. Side winds set up wheel flange friction on the leeward side, and cause a greater resistance than head winds.

Miscellaneous resistance cannot be determined, but Sir John A. F. Aspinall, in his Paper on "Train Resistance," read before the Institution of Civil Engineers, November 26th, 1901, page 31, points out that flange friction, oscillation, concussion, etc., are responsible for about 50 per cent. of the total resistance of the train at 80 miles per hour.

From these remarks it will be gathered how numerous and complex are the components making up train resistance, and, therefore, in order to avoid complications when estimating train loads, it is preferable to employ a general equation which expresses the resultant of all the components as some function of the speed. The earliest equations are those of D. K. Clarke :—

$$R = \frac{V^2}{171} + 8$$

$$R_1 = \frac{V^2}{240} + 6$$

where R equals resistance (lbs. per ton) of engine, tender and train, and R_1 resistance of train only, the resultants increased one half for strong side and head winds in combination with curves under one mile radius.

All the features in the above two equations, excepting speed, are constant, but any equation which does not include a factor for the weight of the vehicle is incomplete, as wagons of low capacity offer a greater resistance than high capacity wagons, and so the Author prefers to use a formula which includes the gross weight of the vehicle.

Fig. 3 shows traction resistance, equations and curves of four authorities, Wellington, Chanute, Harding, and Fry. Each equation embodies speed and weight, but Harding, in addition, employs a factor for the area of the front end. A resistance curve based on actual tests is also shown, and this somewhat follows the curve of Lawford Fry. As Fry's curve is based on modern stock, and probably on numerous investigations and experiments, the Author has employed it in investigating and estimating train loads.

GRADE RESISTANCE.—Grade resistance is the one factor susceptible to accurate calculation. In ascending a grade the resistance is increased due to gravity, and can be determined thus :—Suppose the grade is 1ft. per mile, or 1ft. in 5,280, then the pull necessary to lift 1 ton, or 2,240lbs. will be

$$\frac{2{,}240}{5{,}280} = 0.4240\text{lb.}$$

therefore resistance to gravity becomes

$$\frac{2{,}240}{\text{rate of grade}} = \text{lbs. per ton}$$

The resistance for various grades is shown by Fig. 4.

CURVE RESISTANCE.—Curve resistance, which increases inversely to the radius, is an uncertain quantity, but the American Master Mechanics' Association recommends 0.7lbs. per ton degree of curvature for trains, and 1.4lbs. per ton for locomotives. The degree of curvature is found approximately by dividing the radius in feet into 5,730, thus a curve of 10 chain radius is $\frac{5,730}{660} = 8.68$ degrees. Further explanation of the degree of curvature will be found in Molesworth's Pocket Book—Railway Curves, American Practice. The resistance for various curves is graphically shown by Fig. 5.

Based on the foregoing data, the load curves shown by Figs. 6A, 6B, 7A, and 7B, have been plotted. Figs. 6A and 6B indicate loads the two locomotives can start on various grades, on a straight track, and on curves of 6, 8 and 10 chain radius. The loadings for speeds of 15, 20 and 25 miles per hour are shown by Figs. 7A and 7B, these being the speeds adopted in rating *L* and *Y* locomotives. The loads are arrived at in the following manner.

Example 0-8-0 engine hauling on an up grade of 1 in 100 at 15 miles per hour.

Weight of engine	53.75	tons
Weight of tender	42.00	,,
Total	95.75	,,

Piston speed = 405 feet per minute.

Mean effective pressure, Fig. 1, 54 per cent.

$$\text{Tractive effort} = \frac{20 \times 20 \times 26 \times 180 \times 54}{54 \times 100} = 18,720\text{lbs.}$$

Internal resistance of engine = 18lbs. per ton, Fig. 2.

Absorbed by engine $53.75 \times 18 = 967$lbs.

Available tractive force for hauling engine, tender and train = 18,720 — 967 = 17,753lbs. Fig. 3.

Train resistance = 13.5lbs.

$$\text{Grade resistance} = \frac{2,240}{100} = 22.4\text{lbs.}$$

Effort absorbed by engine, tender (as a vehicle) $= 95.75 \times (13.5 \quad 22.4) = 3,437$lbs.

$$\text{Load hauled behind drawbar} = \frac{17,753 \quad 3,437}{13.5 \quad 22.4} = 399 \text{ tons.}$$

The following is a comparative table showing the theoretical loads hauled at 15, 20 and 25 miles per hour, by the two engines on various grades, based on the suggested formulæ of this Paper, and compared with the Lancashire and Yorkshire Railway classified loadings :—

At 15 miles per hour.

Locality.	Up grade.	0-8-0 Type. Classified.	0-8-0 Type. Theoretical.	0-6-0 Type Classified.	0-6-0 Type Theoretical.
Leaving Manchester, Rochdale direction.	1 in 65	330	274	220	225
Leaving Bolton, Blackburn direction.	1 in 72	440	302	280	247
Up Hoghton Bank.	1 in 100	580	399	360	324

At 20 miles per hour.

Locality.	Up grade.	0-8-0 Type. Classified.	0-8-0 Type. Theoretical.	0-6-0 Type Classified.	0-6-0 Type Theoretical.
Leaving Manchester, Rochdale direction.	1 in 65	280	199	170	173
Leaving Bolton, Blackburn direction.	1 in 72	370	219	250	191
Up Hoghton Bank.	1 in 100	480	296	330	252

At 25 miles per hour.

Locality.	Up grade.	0-8-0 Type. Classified.	0-8-0 Type. Theoretical.	0-6-0 Type Classified.	0-6-0 Type Theoretical.
Leaving Manchester, Rochdale direction.	1 in 65	220	137	140	137
Leaving Bolton, Blackburn direction.	1 in 72	290	158	230	152
Up Hoghton Bank.	1 in 100	280	215	280	203

The $10\frac{1}{2}$-ton gross weight wagon is higher than the average goods train-load running on British lines, which is about $8\frac{1}{2}$ tons gross weight per wagon. The higher figure, however, was the average weight of the wagons comprising the trains on which the experiments were conducted. In estimating the trains' loads the closest co-operation and interchange of views of both Traffic and Locomotive Departments is essential, as train resistance is a complex problem. It cannot be estimated simply on the number of wagons, nor upon the loads to be hauled, but it depends upon the number of wagons in which the weight is concentrated. The same weight of empties cannot be hauled as fully loaded wagons. As the number of vehicles is a governing factor, it is important that the weight carried should be crowded in as few a number of wagons as possible. Resistance per ton moved diminishes with the increased gross weight of vehicle. Tests carried out in America on three 17-ton wagons, and one 51-ton wagon, showed that not only was there a saving in the dead weight hauled, but there was also a saving of over 43 per cent. in the demand made on the engine when the weight was concentrated in the one wagon.

Crowding the weight carried in as few wagons as possible is important, and the larger the wagon the better is the paying load.

The employment of high capacity wagons has the following advantages :—

Reduction in length of trains.
More trains per section.
Less siding accommodation.
Less number of shunts.
Less time occupied in each section.

Having dealt with the cylinder powers, we will now proceed to a consideration of boiler power in relation to cylinder power.

BOILER POWER.

Locomotive boiler power may be studied from several aspects.

In the early days of locomotive construction, designers paid very little heed to boiler proportions, for so long as there was ample cylinder power, and sufficient adhesion, it was expected that the prime mover would be capable of performing its duties.

With increasing weights of rolling stock, and higher speeds, the economical limits of the boiler became taxed, resulting in the locomotive running what is sometimes termed ' out of breath.'' It then was realised that the boiler power was not sufficient, with the result that it began to be increased in dimensions, but not in a rational manner, as but little information was available in regard to correct proportions of grates and heating surfaces.

The Lancashire and Yorkshire Railway Co., in 1891, built twenty '' passenger '' engines of the 4-4-0 type; ten of these had 18in. cylinders, and ten 19in. cylinders; in all other respects they were identical. In service, however, the 19in. cylinder engines were not as successful as the 18in. class, owing to demand being greater than supply. The total heating surface of the boiler was 1,216 sq. ft. and the grate area 18¾ sq. ft. As a consequence, the cylinders were reduced to 18in. diameter.

Again, the 2-4-2 type Radial Tank engines of this Company with 17½in. diameter cylinders, which have the same boiler capacity as the aforesaid passenger engines, are more efficient than identical Radial Tank engines fitted with 18in. cylinders.

On the Midland Railway, the 19½in. diameter cylinders of Mr. Johnson's 4-2-2 passenger engines also had to be reduced, as the boiler was found to be too small for the demand. These engines had a total boiler heating surface of 1,217 sq. ft. and a grate area of 24.5 sq. ft.

Mr. Ivatt, when appointed the Chief Mechanical Engineer of the Great Northern Railway, considerably increased on several occasions the dimensions of the boilers of new engines that he built.

In 1898, Sir John A. F. Aspinall made a radical departure in locomotive practice by building a number of 4-4-2 " Atlantic " type passenger engines with boilers having heating surfaces 70 per cent. in excess of those in service at the time, and grate areas 38 per cent. greater.

Mr. Hughes made a further advance in 1908 in the design of the 4-6-0 passenger engines, which had boilers with a heating surface 22 per cent. greater than the above " Atlantic " type.

A few general remarks on boiler power, particularly with reference to the two engines and practice at Horwich, may be of interest.

Assuming that proportions are correct, no locomotive in this country has failed to fulfil its requirements on the score of excessive boiler power, and it is a wise policy to make the boiler as large as axle loads and wheel diameters will permit.

Boiler power is influenced by the quality of coal, rate of combustion, rate of firing, heat absorbing efficiency, and loss due to radiation.

Good coal has a calorific value of as high as 15,000 B.T.U.'s per lb., and poor coal as low as 10,000 B.T.U.'s.

The calorific values and compositions of coals generally used by the Lancashire and Yorkshire Railway Company are as follow :—

	Percentage.					
	Fixed Carbon.	Volatile Matter.	Ash.	Total.	Sulphur.	B.T.U.
North Cawber ...	63.40	31.00	5.60	100	1.75	13,609
Woolley	59.00	32.00	9.00	100	2.75	13,403
Barnboro' ...	62.30	30.90	6.80	100	1.29	13,609
Hemsworth ...	65.90	32.05	2.05	100	1.11	14,006
Hickleton ...	58.90	35.85	5.25	100	2.10	13,621
Houghton ...	64.40	31.65	3.95	100	1.59	14,034
Hodroyd	56.65	35.50	7.85	100	3.01	13,403
Park Hill... ...	64.45	32.00	3.55	100	1.23	13,668
Emley Moor ...	63.95	32.15	3.90	100	1.58	13,622

The different grades for a particular quality of coal adopted by the Baldwin Locomotive Company, as generally satisfactory (see "Locomotive Data"), are as follow:—

Coal with less than 7.5 per cent. volatile matter in the combustible, anthracite.

Coal with 7.5 to 12.5 per cent. volatile matter in the combustible, semi-anthracite.

Coal with 12.5 to 25 per cent. volatile matter in the combustible, bituminous.

Coal with more than 50 per cent. volatile matter in the combustible, lignite.

Anthracite coal contains a high percentage of fixed carbon, and a low percentage of volatile matter, and its heating value per lb. of combustible ranges from 14,600 to 14,800 B.T.U.'s.

For the reason that anthracite burns slowly, and packs closely together, a comparatively large grate and a thinly spread fire are essential. Bituminous and semi-bituminous coals contain a high percentage of volatile matter, and ample firebox volume is necessary for complete combustion. Lignite coal has a woody structure, and burns quickly, its heating value is low, also it is low in fixed carbon, consequently a large grate is essential, as large quantities have to be dealt with.

The rate of combustion is dependent on draught conditions. Experiments carried out on the Lancashire and Yorkshire Railway go to show a vacuum of 5in. of water in the smokebox, when running at a cut-off of 51 per cent., and 3in. with 39 per cent.

Professor Goss, in 1900, announced a formula for relation between smokebox vacuum and rate of combustion as follows:—For bituminous coal, the draught $= 0.037 \times$ lbs. of coal burnt per sq. ft. of grate surface per hour.

Assuming a coal consumption of 100lbs. per sq. ft. of grate surface, which is an average figure for goods train haulage, we get $0.037 \times 100 = 3.7$ inches of water at the smokebox, and with 5in. of vacuum, the rate of combustion becomes $\frac{5}{0.037} = 135$lbs. of coal consumed per sq. ft. of grate surface per hour, which is about an extreme economical limit.

It is possible, however, to maintain a maximum hourly rate of combustion of 200lbs. per sq. ft. of grate area, but evaporation diminishes with increasing rates of combustion, as the following extract from A. E. Johnson's article on "A Consideration of Express Locomotive Design" ("Engineer," August 29th, 1913) indicates :—

Lbs. of coal per sq. ft. of grate per hour.	Lbs. of water evaporated from and at 212°F. per lb. of coal.
200	6.1
180	6.3
160	6.5
140	6.8
120	7.1
100	7.5
80	8.1
60	9.0

The proportion of grate to heating surface is one of importance, and a good ratio is about 1 to 65, but it is impossible to attain this in large boilers with narrow fireboxes without resorting to an extremely long box, which is objectionable, as it taxes the endurance of the fireman.

A 10ft. long firebox is about the maximum for convenience of firing, giving a grate area of 31.5 sq. ft. with water spaces of 2¾in.

The ratios of grate to heating surface of the two engines under consideration are :—

0-8-0	0-6-0
1 to 78.2	1 to 64.8

The air openings between the grate bars is also important as influencing steaming qualities. Information is wanted on this point.

The first engines constructed at Horwich had ⅝in. openings between the grate bars, equal to 25 per cent. of the grate area, giving a proportion of air space to total heating surface of 1 to 313 for the 0-8-0 engine, and 1 to 260 for the 0-6-0 class. The steaming under this condition was unsatisfactory, and therefore the proportion was increased to 33 per cent., with beneficial results. The air openings are now 1 to 237, and 1 to 197, respectively.

The air space through the tubes of the 0-8-0 engine is 4.15 sq. ft., and of the 0-6-0 engine 2.7 sq. ft., giving ratios of tube air space to grate air space of 1 to 2.07 and 1 to 2.3 respectively.

Tube Spacing.—Tendency of late has been to increase the spaces between the tubes, and to give a greater distance between the tube and barrel of the boiler. The distance between the tubes of the first lot of engines built at Horwich was $\frac{9}{16}$in., later this was increased to $\frac{11}{16}$in., then to $\frac{13}{16}$in., and latterly to $\frac{15}{16}$in. The less number of tubes has not been detrimental to the steaming qualities of the engines.

Length of Tubes.—The distance between tube plates of the 0-8-0 engine is 15ft., and of the 0-6-0 engine 10ft. $9\frac{3}{8}$in., the external diameters of the tubes being 2in. and $1\frac{3}{4}$in., giving ratios of diameter to length of 90 and 74 respectively.

In order to determine the most economical tube length some 25 years ago, M. Henri, of the Paris, Lyons and Mediterranean Railway, carried out a number of tests, and found that with a 3in. vacuum the best length appeared to be about 14ft. Some of Henri's experiments were conducted on two goods engines, each having a grate area of 25.5 sq. ft. One had a heating surface of 1,601 sq. ft., and the other 1,756 sq. ft. In the first there were 247 tubes $1\frac{7}{10}$in. external diameter, the other 307 tubes $1\frac{5}{10}$in. external diameter, both being 14ft. long. The investigator was of the opinion that the larger tubes should be preferred. The diameter to length ratio of the larger tube was 1 to 99. The vacuum of 3in. would appear low as compared with that generally obtained now, and with higher vacuums equally good results could be obtained with longer tubes. There would seem to be a good opportunity here for experiment in order to determine the important question of tube length.

Firebox Water Spaces.—The fireboxes of the original Lancashire and Yorkshire engines had $2\frac{1}{2}$in. water spaces, but in 1904 Mr. George Hughes increased this to 4in., and all new boilers have the wider water space. The increased spaces have resulted in a higher mileage and fewer repairs.

Pitching of Stays.—The general practice has been to pitch the stays 4in. apart, each 1in. diameter stay supporting 16 sq. in. Recently the stress in each stay has been reduced, the supporting area now being 13.14 sq. in., giving a stress of 1.05 tons per sq. in.

Measurement of Heating Surface.—There is no generally accepted rule for the measurement of heating surface. It is the usual practice in this country to calculate it on the water side, but the Continental practice is to express the measurement based on surfaces in contact with the hot gases.

The heating efficiency of the firebox is considerably higher than tube efficiency, and tube efficiency is greater at the firebox than at the smokebox end, yet the values are always expressed as one.

The evaporation values of different portions of the heating surface was experimented with on the Northern Railway of France in 1874 (see "Locomotive Diary"). The boiler on which the experiments were conducted had a firebox heating surface of 60.28 sq. ft., and a tube surface of 732.1 sq. ft., giving a total heating surface of 792.43 sq. ft.

For the purpose of the above tests the tubes, 12ft. 3in. long, were compartmented in five sections, four of which were 3ft., and one 3in., the latter being included in the firebox surface.

Below is a table of the results obtained :—

Surface.	Fire-box.	1st section.	2nd section.	3rd section.	4th section.	Total.
Water evaporated in lbs. per sq. ft. per hour.	36.9	11.44	5.72	3.52	2.31	8.9
Water evaporated per section, lbs. per hour.	2820	2047	1024	630	414	6935
Per cent. of total evaporation per section.	40	30	15	9	6	100

This experiment is interesting as it indicates the rapidity with which the efficiency of the tube decreases towards the smokebox end, the last compartment recording only 6 per cent. The Author finds in plotting the curve of percentages that the evaporative value of the tube has entirely disappeared at the length of 15ft. from the firebox end; it must be noted, however, that the rate of combustion is not high. Assuming a ratio of 1 to 65 grate to heating surface the rate of combustion is

$$\frac{6935 \times 65}{8.9 \times 792.4} = 63\text{lbs. per sq. ft. of grate surface per hour.}$$

SMOKEBOXES.—The economical rate of combustion largely depends on the design and proportion of the smokebox, and position of the blast nozzle relative to chimney and tubes. In the original design of the 0-8-0 class, the chimney with $12\frac{1}{2}$in. diameter choke, had an extension which penetrated the smokebox 15in. ; the extension carried a hood. In combination with this a short blast pipe was employed. Later, the chimney was replaced by one of 16in. diameter, with a 2in. extension only; the hood was discarded, and a longer blast pipe used. This alteration resulted in the employment of a $5\frac{3}{4}$in. diameter nozzle, as compared with a

5in. previously used, without any detrimental effects as regards steaming.

LONG VERSUS SHORT SMOKEBOXES.—Mr. George Hughes, in his Paper, "Locomotives Designed and Built at Horwich," remarks as follows:—

> "Investigations tend to prove that the extended smokebox serves as a reservoir, thus assisting the maintenance of draught between each exhaust, and so modifying the intermittent character of the blast."

Having in a general way dealt with boiler practice, we will now proceed to a consideration as to whether the boilers of the two engines referred to are capable of supplying the necessary steam for their respective cylinders, based on an assumed rate of combustion.

The leading particulars of the boilers are given in the following table:—

	0-8-0	0-6-0
Mean diameter of barrel	4ft. 10in.	4ft. 2in.
Length between tube plates	15ft. 0in.	10ft. 9⅝in
Length outside fire-box	8ft. 1in.	6ft. 0in.
Type of fire-box	Belpaire.	Round top.
Number of tubes	239	220
External diameter of tubes...	2in.	1¾in.
Heating surface of tubes in sq. ft. (water side)...	1,877	1,108.73
Heating surface of fire-box, ditto	161.6	107.68
Total heating surface ditto...	2,038.6	1,216.41
Grate area	26.05	18.75
Steam capacity, cu. ft.	76	43
Water capacity, gals.	1,200	740
Ratio of total heating surface to gallons of water	1.7 to 1	1.6 to 1
Ratio of tube air space to grate air space ...	1 to 2.07	1 to 2.3
Ratio of fire-box heating surface to grate area ...	6.2 to 1	5.74 to 1
Ratio of total heating surface to grate area ...	78.2 to 1	64.9 to 1
Tube heating surface to grate area	74 to 1	59 to 1
Fire-box volume, cu. ft.	144.25	99.5

RATE OF COMBUSTION.—According to A. E. Johnson, the average rate of combustion for British express work in heavy service is about 100lbs. per sq. ft. of grate surface per hour, with a maximum of 160lbs. for an engine having 6ft. 6in. driving wheels, and cylinders 19in. × 26in., when running at 54 miles per hour.

$$\frac{54 \times 5{,}280 \times 2.166 \times 2}{60 \times 6.5 \times 3.1416} = 1{,}995\text{ft.}$$

Assuming a mean economical coal burning rate of 130lbs. per sq. ft. of grate surface per hour, for a piston speed of 1,000ft. per minute, the curve indicated by the full line,

Plate VI., Fig. 8, has been drawn. The Author has no definite information regarding rates of combustion for various speeds, but he suggests that they follow somewhat the powers developed in the cylinders, and on that assumption a characteristic curve of an indicated horse-power curve has been plotted.

Referring to this curve, it will be observed that the rate of combustion for a 300ft. piston speed is 68lbs., for a 600ft., 108lbs.; and for a 1,000ft. piston speed, is 130lbs. per sq. ft. of grate surface per hour. The curves represented by dotted lines, Fig. 8, indicate the coal burnt per hour by the two respective engines.

Fig 9 shows the rates of combustion per sq. ft. of heating surface per hour, and corresponding rates of evaporation at and from 212° F. Total evaporation = rate of combustion per sq. ft. of heating surface × heating surface × rate of evaporation. In the tables, Plate X., the rates of evaporation are based on a temperature of 100° F., which is probably the average temperature of the delivery water from the injectors.

The curve shown in Fig. 10 represents lbs. of steam consumed per indicated hourse power hour for various piston speeds, based on a large number of indicator diagrams taken on the Lancashire and Yorkshire Railway. The total I.H.P. the boiler will maintain

$$= \frac{\text{Rate of evaporation}}{\text{lbs. of steam per I.H.P. hour.}}$$

Figs. 11A and 11B are comparative curves of indicated horse power the boilers are capable of maintaining, and indicated horse power developed in the cylinders.

The tables A and B are summaries of particulars from which the various curves have been plotted. Referring to Figs. 11A and 11B, it will be noted that the cylinder powers are in excess of boiler powers; this is particularly pronounced in the case of the 0-6-0 engine, and it would appear from these results that the boilers are working at higher rates of combustion than those indicated by Fig. 8.

CONCLUSION.—The Paper is an endeavour, however imperfect and full of shortcomings, to bring about by discussion and interchange of opinions some generally accepted rules for estimating train loads.

Tractive effort, train resistance and boiler proportions are complex problems upon which much doubt exists as to exact values.

Measurement of heating surface as at present expressed is unsatisfactory for comparison.

Correct proportions of grate and tube air spaces, efficient tube length, spacing of tubes, smokebox capacity, location of blast pipe, design of nozzle, chimney proportions, are further problems which require solving by research and experiment.

There is the need in this country for a plant where locomotive laboratory tests could be conducted, and this need has the sympathy of the Author's Chief, Mr. George Hughes, Chief Mechanical Engineer of the Lancashire and Yorkshire Railway.

The National Laboratory at Teddington is somewhat removed from the locomotive industry, and Manchester being an important locomotive building centre, the Author is of opinion that this city is excellently situated for the installation of a locomotive testing plant. Much valuable work has been done on the Purdue testing plant in America, and many problems investigated relating to locomotive performance.

The data presented is not in any way conclusive, and the Author is conscious of this, but the actual loads hauled at the various speeds by the two engines confirm somewhat the reliability of the suggested formulæ.

As before stated, the Paper only deals with saturated steam locomotives; there is much information wanted in respect of locomotives using superheated steam.

THE RELATION OF CYLINDER AND BOILER POWER TO LOCOMOTIVE RATING.

Particulars from which the Various Curves have been Plotted.

TABLE A.

Piston Speed Ft. per Minute.	Coal burned per square foot of grate surface per hour. lbs.	Coal burned per square foot of heating surface per hour. lbs.	Heating surface. Square feet.	Total Coal burned per hour. lbs.	Water evaporation at 100°F. per square foot of heating surface per hour. lbs.	Water evaporated per hour. lbs.	Steam consumption per I.H.P. hour. lbs.	I.H.P. Boiler will Maintain.
"A" 0-8-0 Type Engine.				(Ratio of Grate to Heating Surface 1 to 78.2.)				
300	68	.87	2038.6	1771	8.35	14790	27.35	540
400	87	1.112	,,	2266	7.85	17800	26.00	685
500	98	1.254	,,	2553	7.45	19020	25.25	754
600	108	1.382	,,	2813	7.10	19980	25.00	799
700	117	1.496	,,	3047	6.85	20880	25.25	827
800	122	1.560	,,	3178	6.68	21220	26.00	817
900	127	1.622	,,	3308	6.55	21670	27.35	792
1000	130	1.661	,,	3368	6.50	22100	30.00	734

TABLE B.

"B" 0-6-0 Type Engine.				(Ratio of Grate to Heating Surface 1 to 64.86.)				
200	50	.770	1216.46	937	8.75	8200	30.00	273
300	68	1.049	,,	1295	7.75	10040	27.35	367
400	87	1.342	,,	1631	7.25	11830	26.00	450
500	98	1.512	,,	1837	6.75	12400	25.25	491
600	108	1.664	,,	2025	6.50	13160	25.00	527
700	117	1.802	,,	2193	6.25	13710	25.25	543
800	122	1.880	,,	2289	6.10	13960	26.00	537
900	127	1.958	,,	2381	5.95	14170	27.35	518
1000	130	2.002	,,	2435	5.85	14240	30.00	475

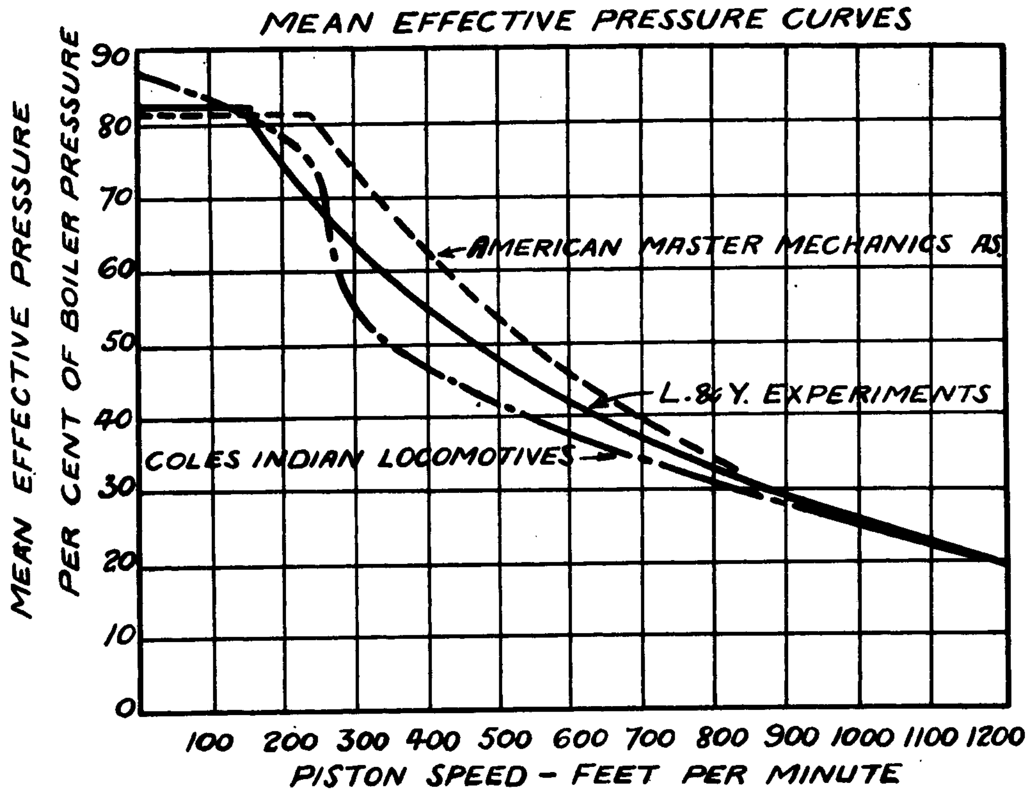

MEAN EFFECTIVE PRESSURE CURVES
MEAN EFFECTIVE PRESSURE
PER CENT OF BOILER PRESSURE
90
80
70
60
50
40
30
20
10
0
AMERICAN MASTER MECHANICS AS.
L.&Y. EXPERIMENTS
COLES INDIAN LOCOMOTIVES
100 200 300 400 500 600 700 800 900 1000 1100 1200
PISTON SPEED - FEET PER MINUTE

FIG. 1.

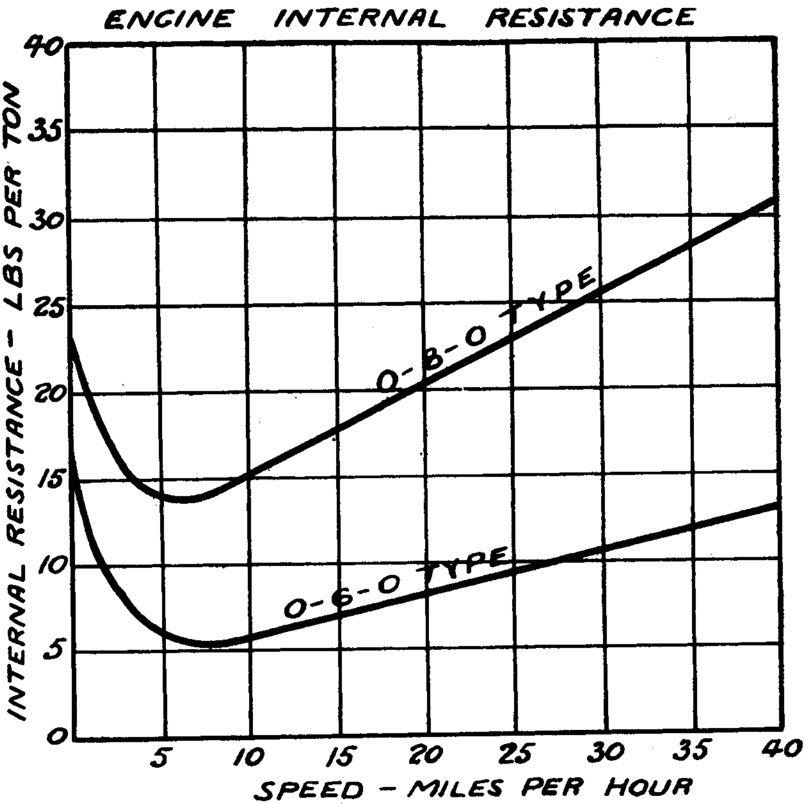
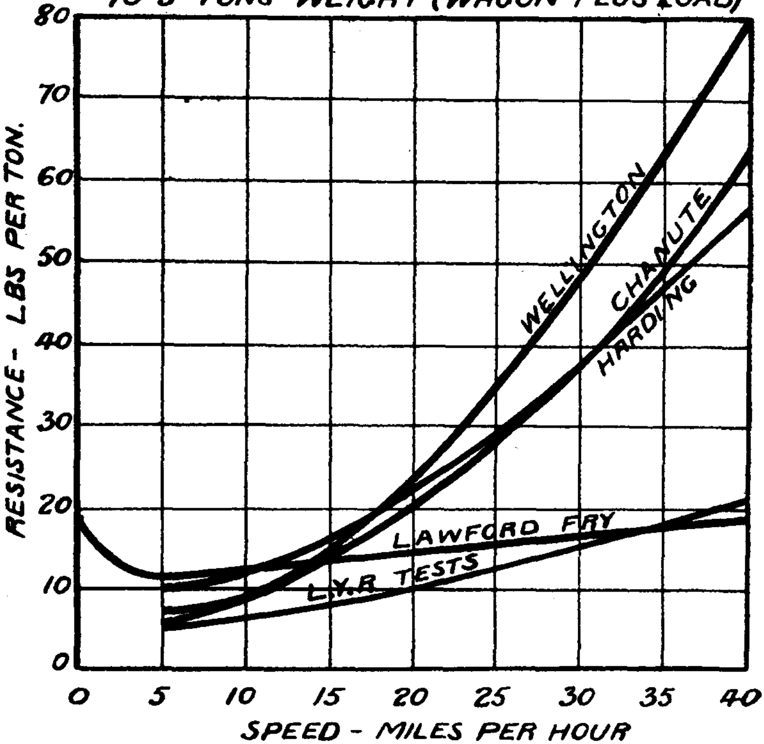

ENGINE INTERNAL RESISTANCE
INTERNAL RESISTANCE - LBS PER TON
40
35
30
25
20
15
10
5
0
5
10
15
20
25
30
35
40
SPEED - MILES PER HOUR
0-8-0 TYPE
0-6-0 TYPE

FIG. 2.

RESISTANCE PER TON OF WAGON OF 10·5 TONS WEIGHT (WAGON PLUS LOAD)



FORMULAE

WELLINGTON	$R = 448 + 0{\cdot}0056\,V^2 + 0{\cdot}46\,\frac{V^2}{W}$
CHANUTE	$R = 56 + 0{\cdot}008\,V^2 + 0{\cdot}3\,\frac{V^2}{W}$
HARDING	$R = 6 + \frac{V}{3} + \frac{0{\cdot}0025\,V^2 A}{W}$
LAWFORD FRY	$R = 1{\cdot}5 + \frac{106 + 2V}{W+1} + 0{\cdot}001\,V^2$

WHERE R = RESISTANCE TO TRACTION IN LBS PER TON
V = VELOCITY IN MILES PER HOUR
W = WEIGHT OF CARRIAGES IN TONS
A = AREA OF FRONT IN SQ FT.

FIG. 3.

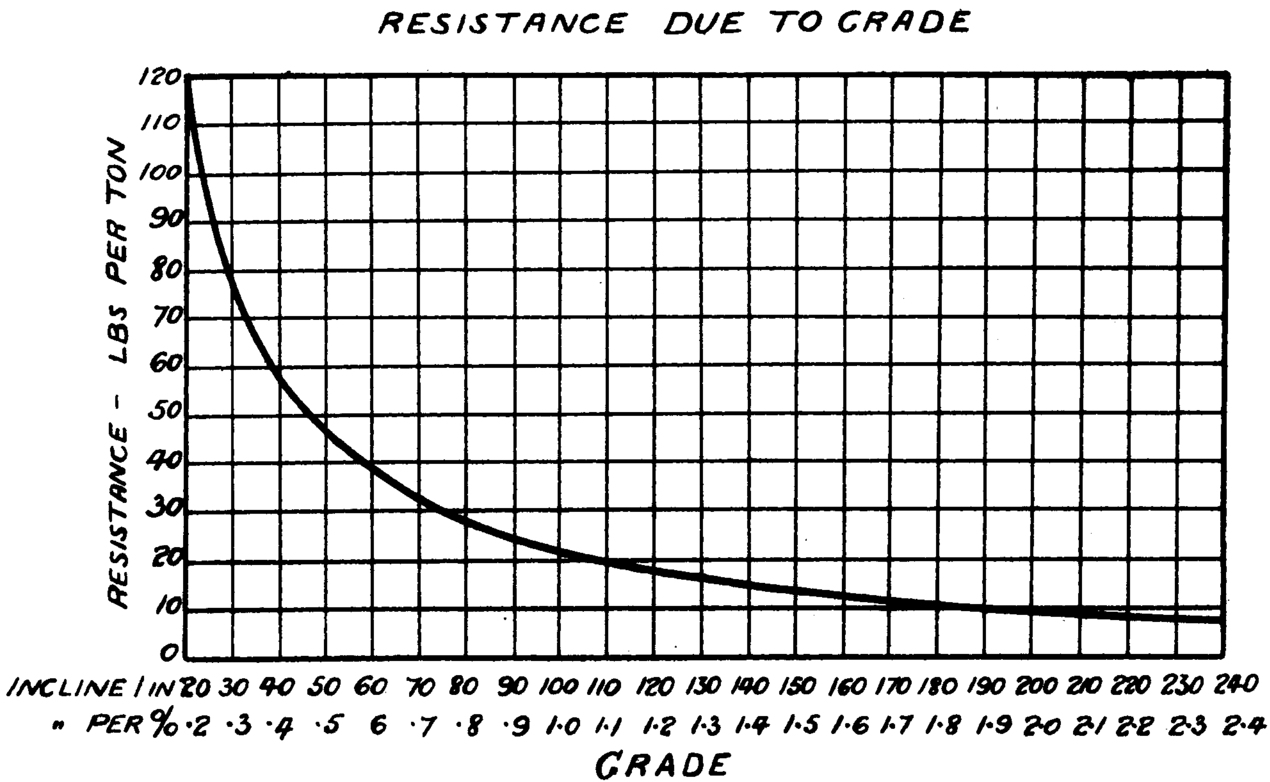

RESISTANCE DUE TO GRADE
RESISTANCE - LBS PER TON
120
110
100
90
80
70
60
50
40
30
20
10
0
INCLINE 1 IN 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210 220 230 240
" PER % ·2 ·3 ·4 ·5 6 ·7 ·8 ·9 1·0 1·1 1·2 1·3 1·4 1·5 1·6 1·7 1·8 1·9 2·0 2·1 2·2 2·3 2·4
GRADE

FIG. 4

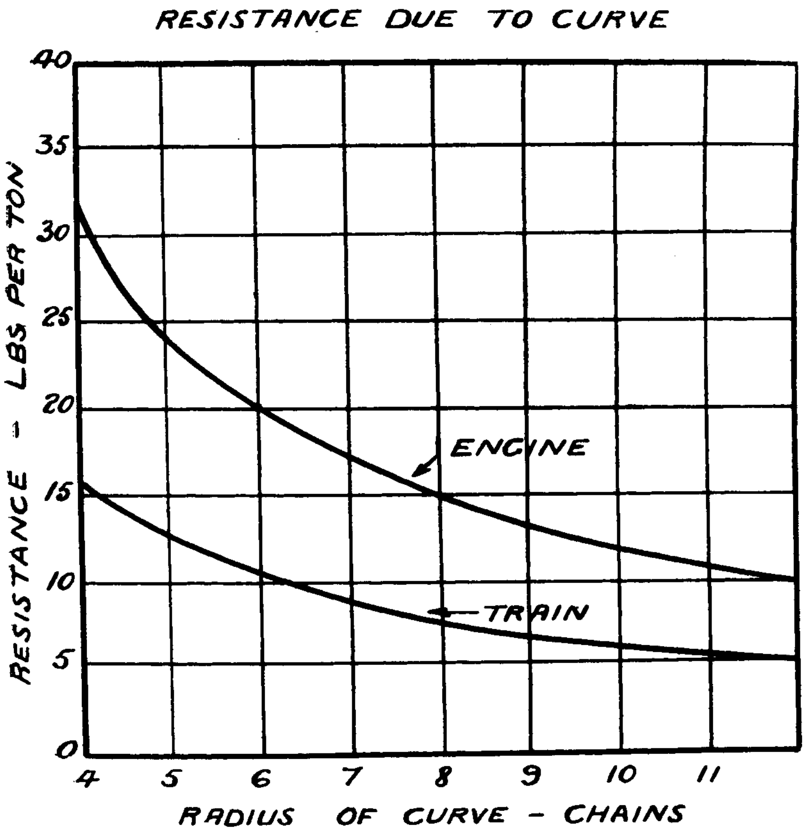

RESISTANCE DUE TO CURVE
40
35
30
25
20
15
10
5
0
RESISTANCE - LBS PER TON
ENGINE
TRAIN
4
5
6
7
8
9
10
11
RADIUS OF CURVE - CHAINS

FIG. 5.

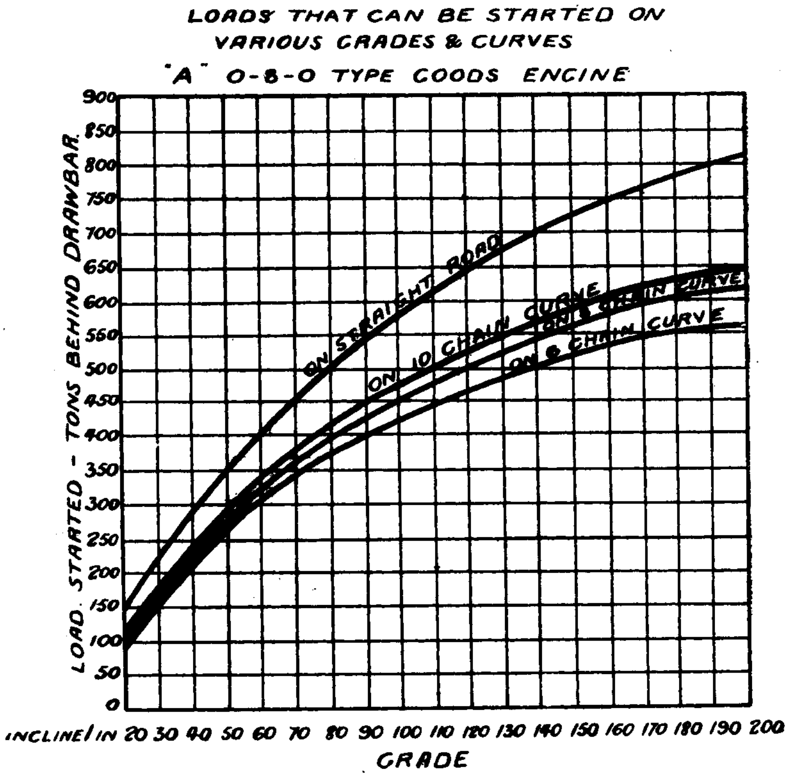

LOADS THAT CAN BE STARTED ON
VARIOUS GRADES & CURVES
"A" O-8-O TYPE GOODS ENGINE
LOAD. STARTED - TONS BEHIND DRAWBAR.
900
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0
ON STRAIGHT ROAD
ON 10 CHAIN CURVE
ON 8 CHAIN CURVE
ON 6 CHAIN CURVE
INCLINE 1 IN 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200
GRADE

FIG 6 A

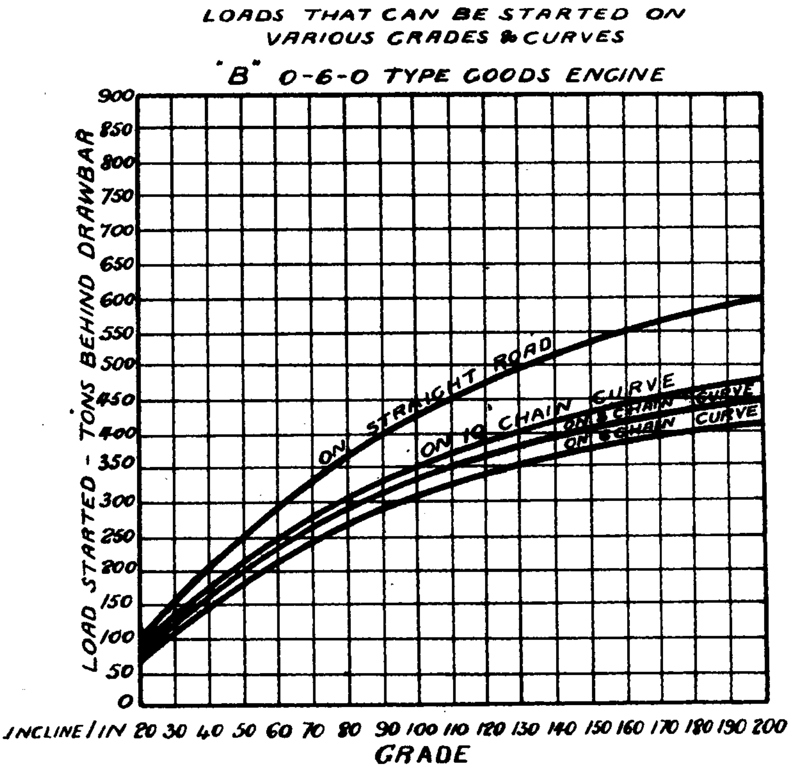

LOADS THAT CAN BE STARTED ON
VARIOUS GRADES & CURVES
"B" 0-6-0 TYPE GOODS ENGINE
LOAD STARTED - TONS BEHIND DRAWBAR
900
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0
ON STRAIGHT ROAD
ON 10 CHAIN CURVE
ON 8 CHAIN CURVE
ON 6 CHAIN CURVE
INCLINE 1 IN 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200
GRADE

FIG. 6. B.

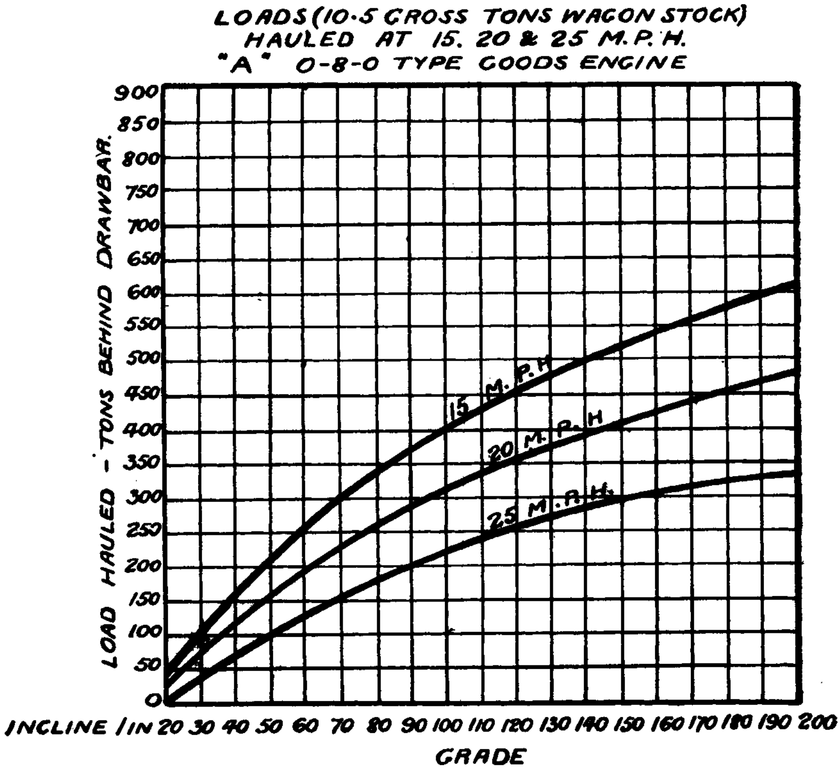

LOADS (10·5 GROSS TONS WAGON STOCK)
HAULED AT 15. 20 & 25 M.P.H.
"A" 0-8-0 TYPE GOODS ENGINE
LOAD HAULED - TONS BEHIND DRAWBAR.
900
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0
15 M.P.H
20 M.P.H
25 M.P.H.
INCLINE 1 IN 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200
GRADE

FIG. 7. A.

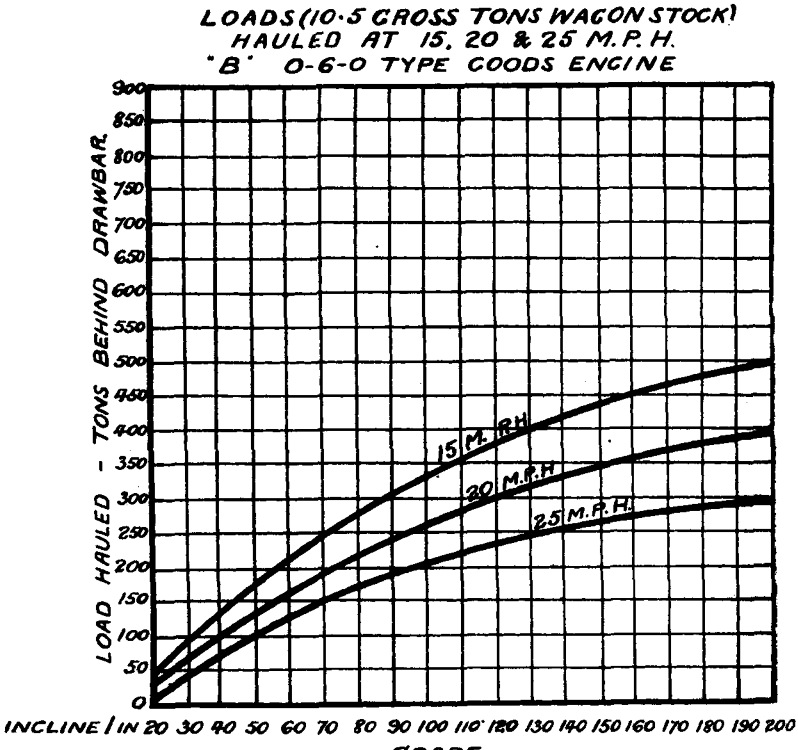
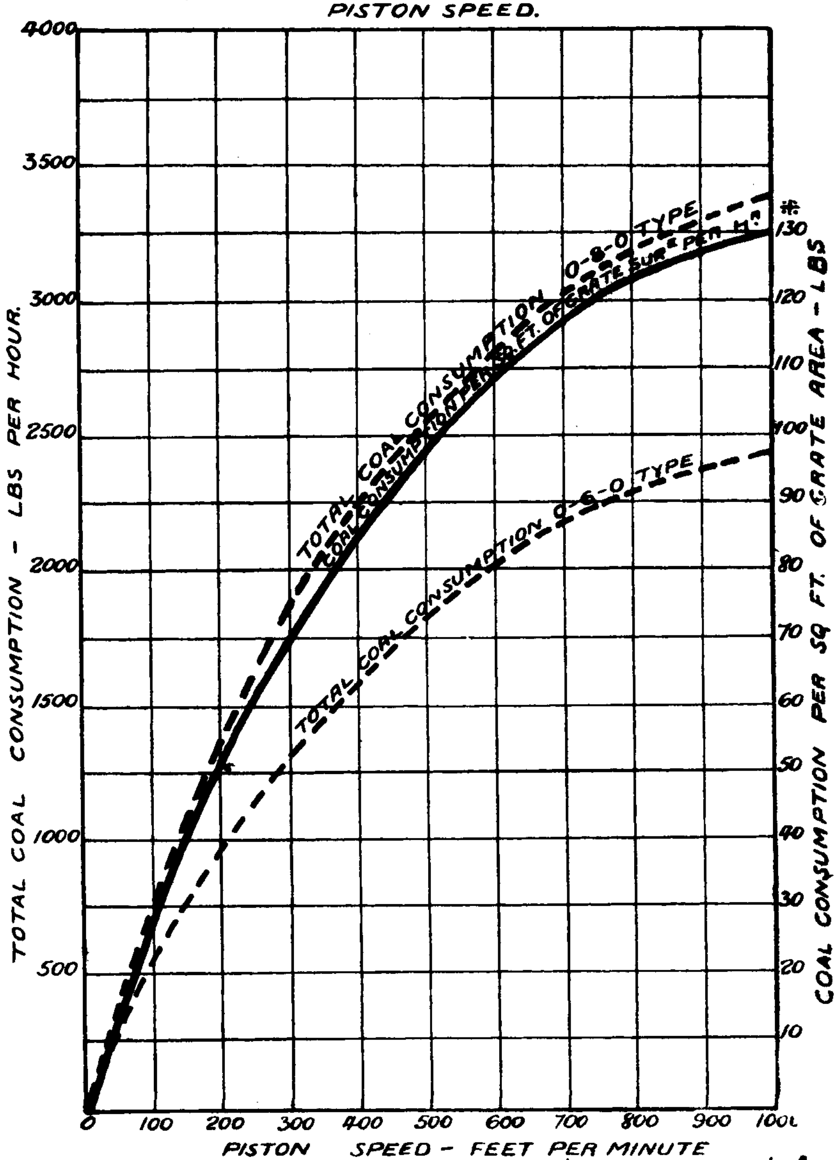

LOADS (10·5 GROSS TONS WAGON STOCK)
HAULED AT 15, 20 & 25 M.P.H.
"B" 0-6-0 TYPE GOODS ENGINE
LOAD HAULED - TONS BEHIND DRAWBAR.
900
850
800
750
700
650
600
550
500
450
400
350
300
250
200
150
100
50
0
15 M.P.H
20 M.P.H
25 M.P.H.
INCLINE 1 IN 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200
GRADE

FIG. 7. B.

TOTAL COAL CONSUMPTION & COAL CONSUMPTION PER SQ. FT OF GRATE AREA IN RELATION TO PISTON SPEED.



*: THIS FIGURE IS BASED ON JOHNSON'S FIGURE FOR 6'-6" PASSR ENGINE RUNNING AT 54 M.P.H. PISTON STROKE 26"

FIG. 8.

COAL CONSUMPTION & EVAPORATION IN RELATION TO PISTON SPEED.

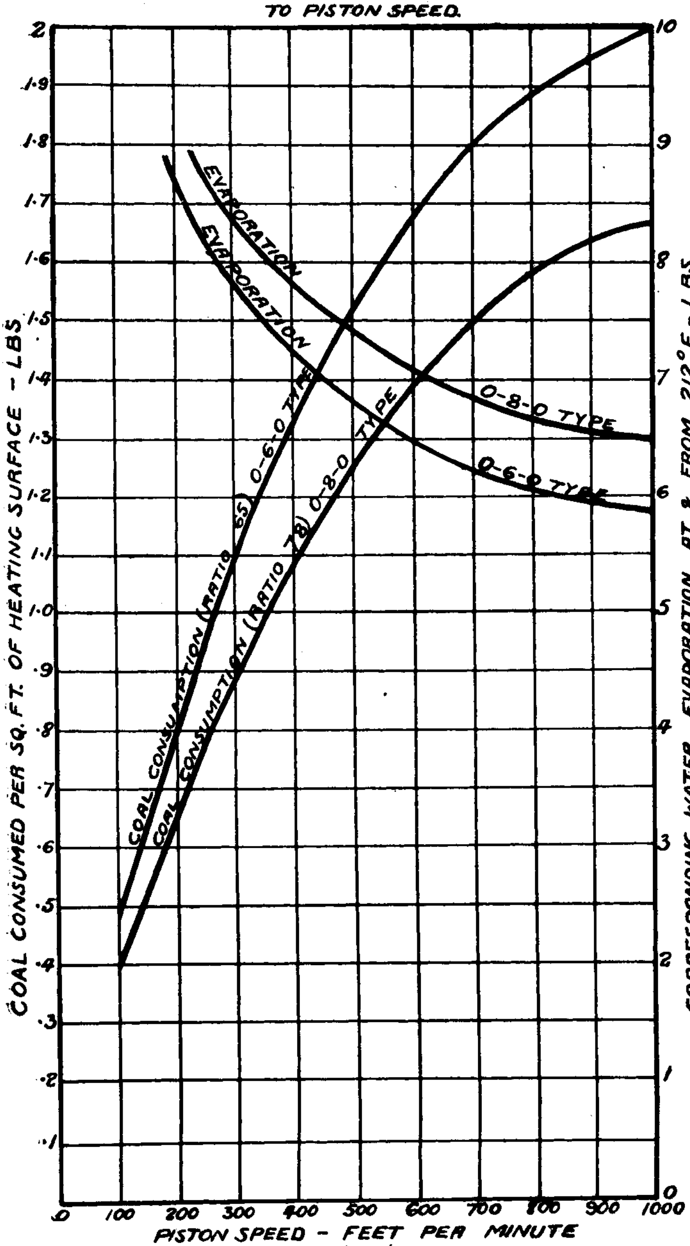


FIG. 9.

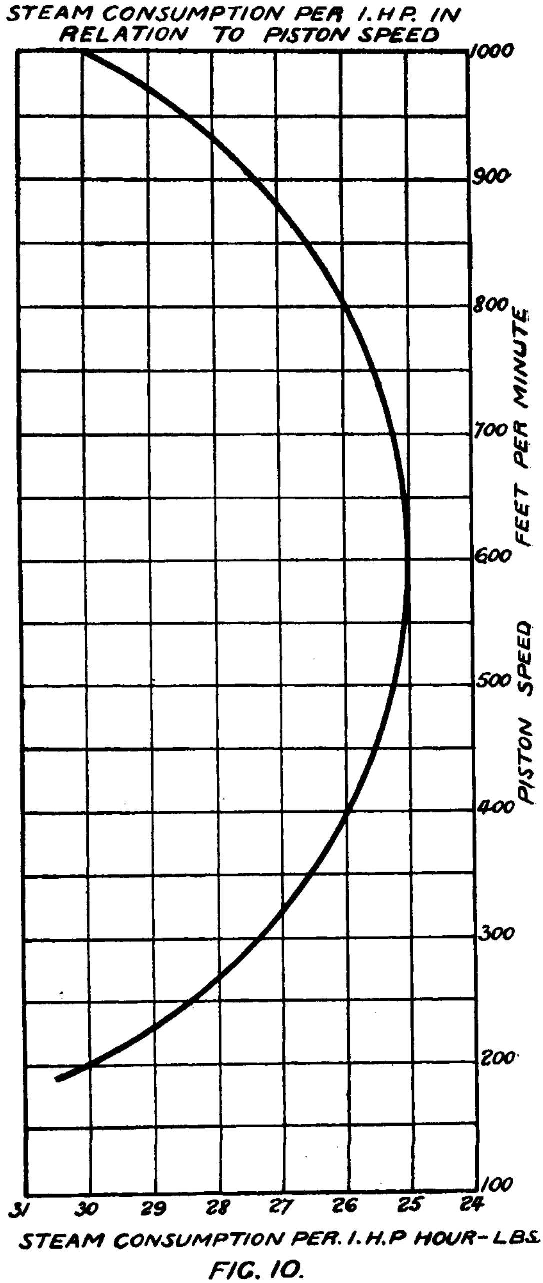

STEAM CONSUMPTION PER I.H.P. IN
RELATION TO PISTON SPEED
1000
900
800
700
600
500
400
300
200
100
PISTON SPEED FEET PER MINUTE
31
30
29
28
27
26
25
24
STEAM CONSUMPTION PER. I.H.P HOUR-LBS.

FIG. 10.

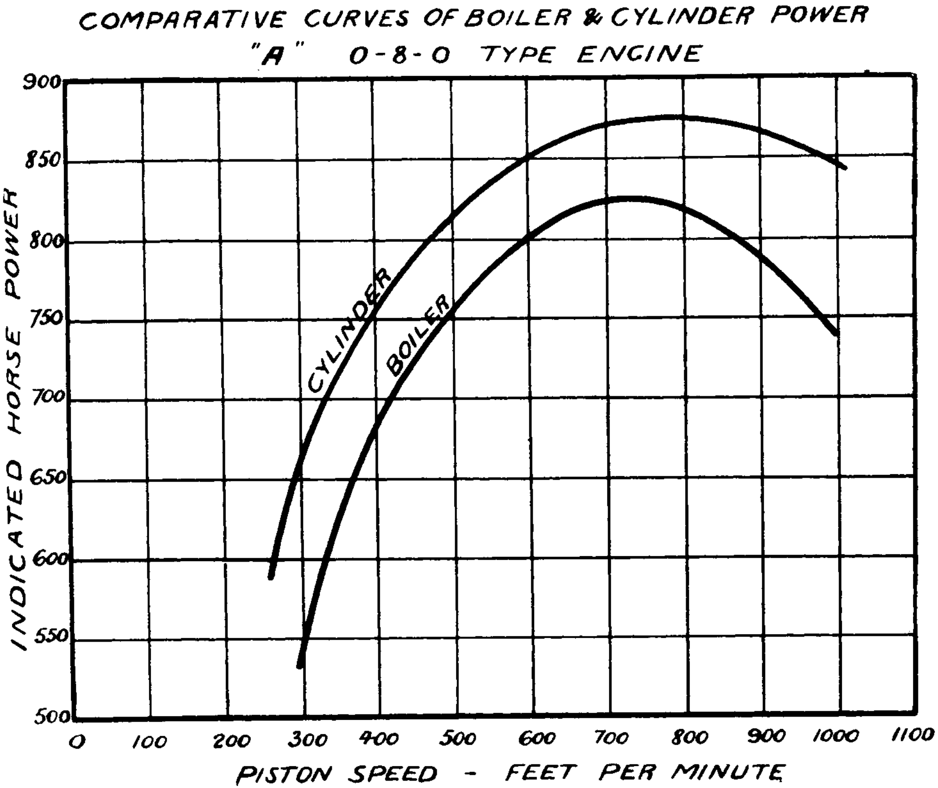

COMPARATIVE CURVES OF BOILER & CYLINDER POWER
"A" 0-8-0 TYPE ENGINE
INDICATED HORSE POWER
900
850
800
750
700
650
600
550
500
CYLINDER
BOILER
0
100
200
300
400
500
600
700
800
900
1000
1100
PISTON SPEED - FEET PER MINUTE

FIG. 11. A.

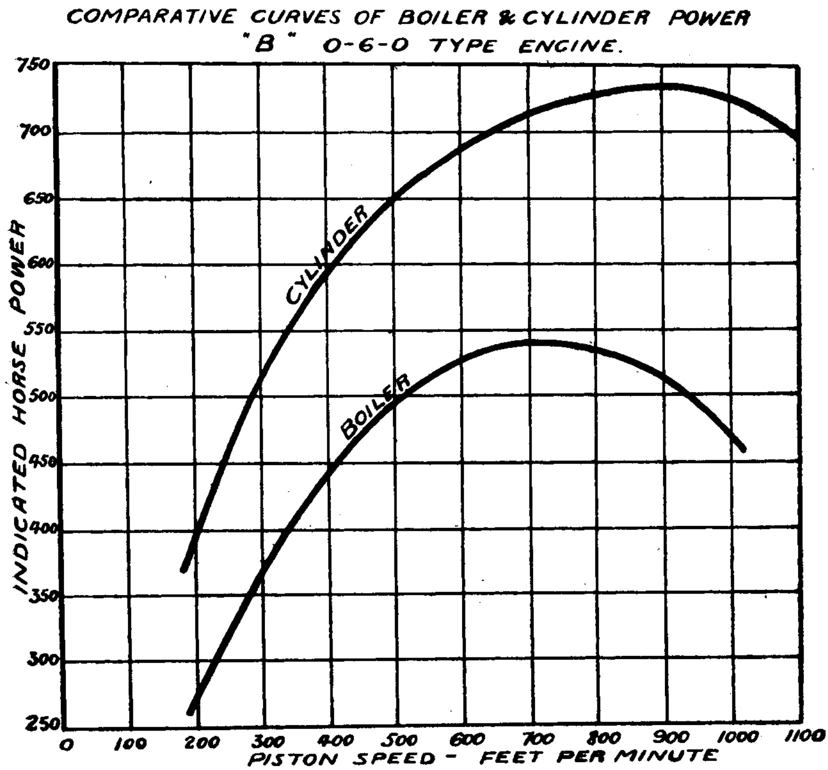

COMPARATIVE CURVES OF BOILER & CYLINDER POWER
"B" 0-6-0 TYPE ENGINE.
INDICATED HORSE POWER
750
700
650
600
550
500
450
400
350
300
250
CYLINDER
BOILER
0
100
200
300
400
500
600
700
800
900
1000
1100
PISTON SPEED - FEET PER MINUTE

FIG. 11.B.

DISCUSSION.

The Chairman: I have been greatly interested in the very excellent Paper dealing with a subject as contentious as any I know. I think that the indebtedness of this meeting is due to Mr. Gass for the pains he has taken to prepare the Paper, which is one that lends itself to discussion from all points of view of a locomotive man, for if he does not know anything about train resistance he knows something in respect to proportions of boilers and cylinders and many other things that Mr. Gass has touched upon. I now move that the best thanks of this meeting be given to Mr. Gass for the very excellent Paper he has presented to this meeting and I ask you to record your appreciation in the usual manner.

As I have already remarked, the Author has touched upon a very contentious subject, and I say this for the reason that there are so many variable conditions which have to be dealt with in considering the rating of locomotives. From experience, I have found that the more experiments one carries out the more are these tantalising varying conditions realised; in fact, I go so far as to say that it is almost impossible to repeat an experiment under actually the same conditions. In going out with a locomotive and train, certain results are obtained; an alteration is made to the engine, after which one may have to wait weeks before the same weather conditions arise as obtained on the first trial and it may be that some of the other variables have altered.

With regard to the establishing of definite formulæ for train resistance, in my opinion, the conditions met with in the operation of railway trains are so variable that it is practically impossible to establish more than a general equation. To establish even general formulæ for train resistance requires a commendable amount of patience, unlimited time and hard work to accomplish. One method of ascertaining train resistance, which was put into operation some years ago, was to turn the engine into its own dynamometer. To do this, after the valves had been most carefully set, both sides absolutely synchronised, indicator diagrams were taken from the engine under various working conditions, from a few miles per hour up to the maximum speed. The points of cut-off, the boiler pressure, position of steam regulator and direction and force of wind were carefully recorded and by these means a chart was produced from which it was possible to plot a continuous horse-power curve for the engine. So far as the particular company's passenger stock, at any rate, was concerned, it was found that from 20 m.p.h. upwards the resistance on the level of the engine and train combined approximated closely to

$$R = 3 + \frac{V^2}{250}$$

Even in putting this formula forward, I am particular to make the same reservation as Mr. Gass that the equation must not be looked upon as final and probably would not be applicable to the stock of all railways. At speeds less than 20 m.p.h. it is very difficult to obtain any reliable data for train resistance, but I will describe briefly more experiments which were conducted to ascertain this, and hope to hear some day that it has been considerably improved upon. A special piece of track was selected and marked off; the engine was started some distance from a point marked "A," where an electric contact was fixed; other contacts were fixed at "B," "C" and "D," the distance between each of the points being exactly the same. Passing through each section, indicator diagrams were taken, the boiler pressure, points of cut-off, position of regulator and time taken to pass from point to point were carefully recorded. The train in these tests consisted of loaded tenders, and it should be noted that the road was absolutely straight and practically level, the small difference from actual level being allowed for in the subsequent calculations. It was found, using the following formula:—

$$R = \frac{375 \text{ h.p.}}{\text{W.V.}}$$

that at 11.29 m.p.h. the resistance for engine and train was 14.6lbs. per ton, where:—

R = resistance in lbs. per ton for gross moving weight on level.
W = weight of train, including engine and tender.
V = indicated horse-power.

I have made these few remarks to emphasise what the Author has pointed out very conclusively at the end of the Paper, that the drawing up of any absolute formula for train resistance, under the many varying conditions met with, is almost an impossible task. It is well known that it is possible to marshal a long goods train in such a way that it will take less pulling by one method than by another. It will be found that by placing the heaviest wagons next the engine, the train is easier to haul than when the heavier wagons are placed at the tail end, that is of course taking into consideration that our railways are not straight. There is one question I would like to ask the Author. He states that the internal resistance of the 0-6-0 type engine equals $16\frac{1}{2}$lbs. per ton, whereas that of the 0-8-0 type is 23lbs. per ton. To what does he attribute the difference? I do not propose to take up any more of your time and now call upon the meeting to discuss the Paper, and I am sure that the Author will be only too pleased to answer any questions.

Mr. Ernest F. Lang (Gorton Foundry): I have listened to the Paper with very great interest and think it a most valuable contribution. As an opening contribution I can imagine no paper better fulfilling its purpose, as it embraces so many of the problems relating to locomotive construction.

The Paper apparently only deals with single expansion locomotives using saturated steam, and one therefore has to remember that all formulæ based on those conditions are altered in the case of compound locomotives, and further, when the steam is superheated the factors again become changed. Thus we may hope that the Author will develop his subject further in future valuable papers.

The suggestion that Manchester is a fitting centre for a locomotive testing plant is an admirable one, and I am quite sure that some day the proposal will be carried out. The Purdue experimental plant in America has won the attention of engineers all over the world, and surely in this great and wealthy country of ours, the seat and origin of locomotive development, it is not too much to ask for funds for the installation of a locomotive testing plant, and our own municipal Education Committee might well father the scheme. I am sure the results obtained from experiments on such a plant would be far-reaching, and important improvements in locomotive construction would follow.

One minor advantage that an experimental plant could give, but which has not yet been tried, has been called to my mind by Mr. Smith's remark as to the effects of weather. An increase in the moisture of the air has more effect on combustion than is often imagined. Some years ago Mr. Gayley, experimenting at Pittsburgh, astonished the whole world of iron manufacture by showing that it would pay to dry the air supplied to a blast furnace by passing the same through a refrigerator, all moisture thus being precipitated.

The inclusion of a small dry-blast apparatus in a locomotive testing plant would enable one to reproduce any desired atmospheric condition. Many of you will probably have noticed how much brighter combustion is on a clear night when a large amount of the moisture in the air has been deposited in the form of dew.

As regards wind resistance, advantages have been claimed for a pointed smokebox door at the higher speeds on some Continental railways; here again the effect of shape could be tested by models exposed to the effect of air currents of varying velocity. The motorist, although well favoured in respect of horse-power, does not disdain the advantage of having a " streamline " body on his car.

Special thanks are due to the Author for the great care and trouble which he must have taken in preparing the many diagrams which co-ordinate and compare a vast amount of valuable information for the purpose of his Paper; they will be studied with great interest.

Before concluding these few remarks, I would like to refer to the account given in the " Engineer " of 21st March, 1919, of M. Herdner's presidential address before the French Society of Civil Engineers. M. Herdner is a railway man and he reviews the history of the express engine in France since the year 1878, and the operation in service of single and compound locomotives under saturated and superheated conditions.

Technical history has been hitherto somewhat neglected, although, as shown by the writings of Gairns and Ahrons, among others, locomotive history is extraordinarily interesting, and it is to be hoped that this subject will also prove a frequent theme for papers before the Institution.

Mr. Rowland (G.C. Rly.): With reference to the curves shown in Fig. 1, the reason for the maximum pressure being maintained up to a higher speed in the case of American engines is the longer valve travel and longer ports generally employed in that country. In this way an improved opening is obtained for the admission of steam and a higher initial pressure maintained up to a higher piston speed. The Author remarks that curve resistance is an uncertain quantity, but I find no difficulty in calculating curve resistance by the following method:—

RESISTANCE ON CURVES.

First.—That due to the different lengths of the inner and outer rails causing one or both wheels of a pair to slip. Let R be the mean radius of the curve. Let G = gauge. W = weight on a pair of wheels. Then the difference between the lengths of the rails in a complete circle of radius R is:—

$$2\pi\left(\frac{R+G}{2}-\frac{R-G}{2}\right)=2\pi G$$

This must be multiplied by the weight on one wheel of the pair and by μ the coefficient of friction at very low speeds, since the speed of slip is small, and divided by the total distance travelled by the vehicle, *i.e.*,

$$\frac{\mu 2\pi G \times W/2}{2\pi R}=\frac{\mu WG}{2R}$$

Second.—The second part is that due to the vehicle making a complete rotation as it passes round a circle of the radius R. In a four-wheeled vehicle this is equal to $W \times l \times 2\pi \times \mu$ in which W is the weight on each pair of wheels and l is the wheelbase, *i.e.*, it is equal to $\mu\pi Wl$ in which W is the total weight of the vehicle. Divide by $2\pi R$ the length travelled by the vehic'e during the rotation $\frac{\mu Wl}{2R}$. Adding these two gives $\frac{\mu W}{2R}(G+l)$ which is Morrison's formula.

Assume a third pair of wheels at half wheelbase with equal weights on each pair. Then the second part of the resistance becomes

$$\frac{\frac{\mu 2W}{3}\times\frac{2\pi l}{2}}{2\pi R}=\frac{\mu Wl}{3R}$$

which is less than before, as the middle axle is radial to the curve. In this case the total resistance is

$$\frac{\mu W}{R}\left(\frac{G}{2}\times\frac{l}{3}\right).$$

The maximum value for μ is about 600lbs. per ton or approximately .25. For a four-wheeled vehicle with 8ft.

wheelbase, gauge $=$ 4ft. $8\frac{1}{2}$in. $R = 5$ chains or 330ft. The resistance is $\frac{.25}{660}(4.9166+8) = \frac{3.2544}{660}$ ton per ton or $\frac{2240 \times 3.2544}{660} = 11.045$lbs. per ton.

The extra resistance due to flange pressure is very small. The total work done against the above resistance is evidently $2\pi R \times 11.045$ ft. lbs. per ton in going round the complete circle and the lateral force at the flanges is therefore 11.045lbs. per ton.

If this were applied through a frictionless medium, such as a wheel on a vertical axis, there would be no work done. Actually it is applied by flange pressure acting at a point about $\frac{1}{2}$in. below the tread line, consequently there is a little slip amounting to $2\pi(r+\frac{1}{2}\text{in.}-r)$ per revolution or 2π 1ft.$/24 = \pi/12$ft. per revolution of the wheels. The total work done is therefore $\mu \times \pi/12 \times 11.045$ ft. lbs. per revolution, and as there are R/r revolutions in the complete circle (or probably $R/r+\frac{1}{4}$in.).

The work done in going round the circle is $R/r\ \mu\pi/12 \times 11.045$ ft. lbs., and as this is exerted over a distance of $2\pi R$ ft., the resistance due to flange friction is $\mu \times 11.045/24r$.

Calling μ .25 and r 2ft. makes this $11.045/192 = .0578$lbs. per ton, so that the total resistance becomes $11.045 + .0578 = 11.1$lbs. per ton. Putting this in general terms gives:—

$$F = \frac{\mu W}{2R}(G+l) + \frac{\frac{\mu W}{2R}(G+l)}{24r}$$

The first term in the second bracket, when multiplied by $\mu W/2r\,(G+l)$, is Morrison's formula, the second is equal to Morrison's formula multiplied by $1/96r$ when μ is .25 and this becomes smaller as r the radius of the wheel is increased. This is obviously so for it really amounts to making the flange shallower in proportion to the diameter of the wheel.

The Author states that the L. and Y.R. practice of late has been to reduce the load on firebox stays. I do not think, however, that the failure of stays is due to excessive load imposed by steam pressure, but rather the breaking is caused by movement of the inside box when under heat. In my opinion it is wrong to make the upper rows of stays (which is usually the practice) larger than the stays nearer the foundation ring as the bending becomes greater the farther the stays are removed from the bottom of the box.

Mr. Haigh (L. and Y. Rly., Bolton) said that Mr. Gass had mentioned many points well worthy of discussion, but that he would refer to one only.

An attempt to overcome air resistance was made by some foreign railway companies in adopting the conical or pointed smokebox, but Mr. Haigh thought that more resistance was offered by the irregular shaped or rough sides of the vehicles than by the front of the engine, and suggested that more attention paid to the design of coaches in this respect would lead to higher efficiency.

Mr. D. Corbett Fletcher (G.N. Rly., Manchester): Mr. Gass does not mention anything about brick arches. Will he give us particulars later on about the length of brick arches in these two engines?

Mr. Holcroft (S.E. and C. Rly., Ashford): The Paper is particularly interesting in that it applies to goods train working rather than passenger, and for this reason is very welcome as it presents the side of locomotive working that is less discussed.

The curves shown in Fig. 2 for internal resistance are apparently obtained from the results of experiments, but they are not consistent with the results to be expected. The minimum resistance of the 0-8-0 and 0-6-0 engines is $13\frac{1}{2}$ and $5\frac{1}{2}$lbs. per ton respectively, the total resistance due to internal friction being $13\frac{1}{2} \times 53.75$ and 5.5×42.15 for the two engines. By multiplying these out the 0-8-0 will be shown to have over three times the resistance of the 0-6-0. On this basis an engine with only two pairs coupled, such as a 4-4-0 engine, would be frictionless. The method of arriving at engine resistance is not stated, but any accurate result is always most difficult to obtain on account of the complicated nature of the problem. The chief trouble lies in the segregation and measurement of the force absorbed in acceleration since a very large pull is required to produce any appreciable increase in velocity, on account of the heavy mass of the engine and train. The ordinary methods of recording speed are not sensitive enough to enable more than an approximate estimate of the acceleration to be made, while the irregular drawbar pull adds complications. For this reason large errors in the resistance may occur, so that a considerable number of observations should be made to get a reliable average. Curves are given for the resistance of goods wagons, but these cannot be relied upon in the same way as for passenger stock which is fairly uniform. Nearly all goods trains are made up of a heterogeneous collection of vehicles with widely varying individual resistances and it is necessary to consider the composition of a train before making any estimate of the hauling power required at various speeds. Trains made up of fairly uniform stock, such as coal trains, etc., can of course be dealt with more easily.

The classified list of loadings shows that the actual loads that can be hauled are in excess of the theoretical. Although the grade is stated, the length of such maximum grade is not given. A short length can either be rushed by momentum or climbed at a lower speed than the rated, if circumstances prevent the former. For this reason the whole nature of the road rather than its maximum grade needs to be considered for rating purposes. With regard to the plea for a testing plant as established at Purdue, such an experimental station already exists at Swindon, fully equipped. It is but little used because practical experience with it shows that the locomotive fails to reproduce the same performance as on the road. Absence of vibration and the irregularities of the permanent way alter the movements of the engine while the rush of air reacting on the ashpan and chimney also is lacking. There is less cooling of hot surfaces in still air so that misleading results may be obtained as to losses in radiation. The coefficient of friction between tyres and supporting rollers of the plant is very low so that heavy drawbar pulls cannot be recorded on account of slipping, while at low speeds the turning is very irregular on account of the absence of any "fly-wheel" action arising from the mass of the moving engine and train. The most reliable way of testing locomotives is on the open road, indicated diagrams being taken in conjunction with an autographic record of the drawbar pull and speed by means of a dynamometer car. Such tests are not infrequently made, but what is wanted is the interchange and publication of these records so that the information obtained can be made available for a wide circle of locomotive engineers and not confined to the private information of a few.

Mr. I. E. Mercer (L. and N.W. Rly., Burton-on-Trent): The question of cylinder volume is a rather complex one in many cases, but whatever the maximum effort required it must be met irrespective of mere normal running conditions. A sharp incline at a starting point or anywhere, including a home signal that is at all frequently "against" trains climbing the bank, may at once decide the size to be used. Where general running conditions alone need be considered wide divergence exists in both practice and theory. I would point out that valve and valve-gear design may play no unimportant part. One has the double aim of working the traffic satisfactorily and of doing it most economically; ample cylinder volume if adequately backed by boiler power will satisfy traffic requirements, but how far should it be modified by requirements of economy? Some contend that 20 per cent. cut-offs should be used for economy, and if so this will all tend to the retention of large cylinders, but personally my experience does not at all support the 20 per cent. idea, and the comparative failure of the large cylinders on L. and Y. and M.R., mentioned in the Paper, would tend to support the belief that mere "notching-up" beyond a certain limit does not increase economy even if it does reduce terminal pressures. After all the real point is the weight of steam used per B.H.P. hour—also wear and tear of machinery. So far as actual results go that have come under my notice, I have concluded that with superheated steam a minimum cut-off of about 30 per cent. with regulator full open is the most economical method of working a simple engine; without superheat some amount of throttling seems beneficial. The remarkable results obtained with the unusually small cylinders introduced by Mr. Ivatt seem to favour the small cylinder. But the economical use of steam is very largely dependent on its control; a trip gear will give economy at a cut-off at which a slide valve gear develops the defects of operation that cause loss of economy. Thus, cylinder size must not be determined until the best methods of operation and control have been decided and trials made, but individual engines fitted experimentally will often give results quite opposite to those obtained by two large batches over several years running.

Theoretically a large cylinder should no more get a boiler "out of breath" than a small one on the same work, as the work being the same, the steam consumption should be the same, the cylinder size only affecting the range of expansion and amount of throttling.

As regards the pitching of stays, is it the bending stress that is the vital value, not the tension?

The blast-pipe and smokebox experiments mentioned are interesting, but I do not think any reliable results have yet been obtained. From the Paper before us I gather that, after all, the experiments originally carried out by Mr. Hughes, the results of which he gave to the Institution of Mechanical Engineers, did not prove entirely trustworthy, for the later results mentioned by Mr. Gass are at variance with those first obtained. Large smokeboxes, that is very large ones, also do not seem so effective as we were at first led to suppose. I suggest that the vacuum reservoir effect can be so pronounced that the frictional effect of the blast on the flue gas is affected.

I am no believer in test plants. Trial and experimental engines give such false results as compared with extended running experience, that I feel no practical reliance whatever could be placed on test plant results. All railwaymen know what the first five or so new engines will do while being carefully watched, or even what an old engine will do when being carefully tested, and what widely different results are obtained by drivers left entirely to themselves. After all, from the main or commercial point of view, what a test locomotive will do is not of much importance; it is what the engines will do in everyday working that is all-important.

MEETING AT MANCHESTER, 2nd MAY, 1919.

The Second Ordinary General Meeting of the Manchester Centre of the Institution of Locomotive Engineers was held at the College of Technology, Manchester, on Friday, 2nd May, 1919, at 7.0 p.m., Mr. F. W. Attock (L. and Y. Rly.), Chairman of the Centre, presiding.

The minutes of the First Ordinary Meeting, held on 14th April, 1919, were read and confirmed.

The Paper by Mr. E. M. Gass, on "The Relation of Cylinder and Boiler Power to Locomotive Rating," was further discussed.

DISCUSSION.

The Chairman (Mr. F. W. Attock): I very much regret having missed the reading of the Paper. Of course I am interested in it from the Running Superintendent's point of view and therefore have endeavoured to regard the subject from that standpoint by considering the practical application of the Author's formulæ. I should welcome a simple equation which would enable us to fix our engine loadings, that is knowing the main particulars of the boilers and cylinders, we could readily arrive at the traffic loads that could be hauled at various speeds for any type of engine. Mr. Gass has gone a long way towards that and evolved a comparatively simple method, but I do not think that actual working loads can be determined on a purely theoretical basis because engine and tender resistances are so variable. For example, internal resistance alone varies in the same type of engine when constructed by different locomotive builders. An engine built by the Great Central or Great Northern Railway would probably not give the same results as a similar type constructed by the L. and Y. Rly. or some other company. Further, I should not like to commit myself to anything beyond the theoretical starting effort equation, as I believe the actual loads that can be hauled by any locomotive is a subject for practical tests.

I have had experience of train loadings based on theory and know it has been necessary in practice to alter the schedules originally got out, very considerably after some years of operation, during which alterations have had to be made from time to time, and so eventually train loads have been arrived at which are apparently fairly accurate, anyway they stand the test of working. When I compare the examples given in the Paper, I find they do not altogether agree with the actual loads these engines are hauling; I therefore think this shows that I am right in contending that by theory alone we cannot arrive at the actual train load and that it is absolutely essential to have dynamometer tests for each type of engine. I should like, in conclusion, to say that I happen to have some knowledge of the experiments from which the Author has based and built up his formulæ, and I know that the tests were carried out most carefully and thoroughly, and I wish to put on record that I have every confidence in the figures quoted—certainly as far as they go they will be both useful and helpful, and the Paper will be a most valuable addition to the Proceedings of this Institution.

Mr. Ernest F. Lang (Gorton Foundry): In response to the invitation to speak again at the next discussion, I wish to avail myself of the privilege.

At our previous meeting I was under the same drawback as everyone then present in not having had an opportunity of perusing the Paper beforehand, and now the opportunity presents itself, it is somewhat difficult to speak in anything more than general terms, as the factors in the equations being based on actual tests, one naturally hesitates to criticise without a full knowledge of the conditions under which they were carried out.

In comparing locomotive designs and performances one is confronted by so many variables that it is next to impossible to form fixed rules. The installation of a locomotive testing plant enables many of the conditions to be made constant and, accordingly, by compiling a large number of tests, certain common factors can be built up and theories formulated.

The testing plant would have to be connected with a railway by means of a siding, so that standard gauge locomotives of any railway company could be brought there for testing. It would thus have the advantage over the Purdue University of being able to conveniently test any locomotive and to compare the relative efficiencies of similar types of locomotives under similar conditions. Innovations in locomotive practice, such as Stumpf's "uni-directional flow" engine, could also be tested.

Another thing suggested by the Paper is that the study of locomotive performances is becoming very much more important owing to the increased cost of fuel, material and labour. In Mr. Geo. Hughes' Paper, read before the Institution of Mechanical Engineers in 1910, the coal cost for the L. and Y. Rly. at the pit was 8s. 4d. per ton. I would like to ask what the price is now.

On comparing modern with early locomotive performances one is able to appreciate the effect on locomotive design produced by the ever-increasing necessity for greater tractive effort; so much power has to be provided for the starting resistance alone. Some of the early locomotives were far from good starters, yet once they got moving their performances were wonderful. I refer particularly to the single drivers of the London and North Western and Great Northern Railways. Considering the weight of those engines, their performances have hardly ever been equalled.

There is no doubt but that the present high cost of fuel will bring to the front feed-water heating, and the ever increasing size of locomotives will introduce automatic stoking, for it overcomes the restrictions to the length of the firebox. Again, the question of powdered fuel, which is making headway in the States, will alter many of the prevailing conditions.

In respect of superheating, it appears possible to lower boiler pressure and still retain equal power as when working under saturated conditions. It is astonishing what a difference 40 or 50lbs. of pressure makes to the life of a boiler.

Again, I think that questions concerning valve gears can only be satisfactorily settled by experiments carried out on a locomotive test plant.

We are certainly far from finality in the designing of the steam locomotive and a good many chapters in its history remain to be written before it is superseded by some other motive power.

In this respect reference might be made to the Garrett locomotive which does away with many of the limitations inseparable from the usual design of locomotive, for example, the boiler being carried on a frame suspended from two steam-driven bogies can be designed regardless of width of engine frame and size of wheels; the cylinder dimensions and tractive effort are also two other considerations removed from the usual trammels.

Mr. W. Rowland (G.C. Rly., Gorton): With regard to the heating surface factor I threw this overboard years ago, as the experiments of Isherwood show very conclusively that 80 per cent. of the tubes of a boiler can be stopped up without affecting the heat-absorbing efficiency of a boiler. Lawford Fry's Paper to the Institution of Mechanical Engineers, reprinted in "Engineering," Vol.

LXXXVII., is, I think, a perfectly conclusive proof that the falling off in efficiency as the rate of firing is increased is not in the least due to failure of the boiler to absorb heat presented to it, but simply due to failure of the fuel to burn completely and so develop the heat for the boiler to absorb. I have carried on the work a stage further and think I have succeeded in establishing a satisfactory semi-rational equation governing the efficiency of combustion in any firebox at any rate of firing. In working out the constants of the equation I used the published results of the St. Louis tests as well as Goss' Locomotive Performance and one or two others published by the Illinois University, and as far as I could, I checked them from our own rather imperfect trials. Road tests are as a rule of a very rough and ready nature and of little service for exact data as the Author knows very well. What we want is a good testing plant on the lines of Purdue, and that is a strong point in this excellent Paper. The results of my formulæ applied to the two goods engines referred to are as follows:—

0-8-0 Goods Engine.

Grate area 26.05 sq. ft. × 130lbs. per sq. ft. per hour = 3386.5lbs. per hour, total coal.

Firebox volume = 144.25 cub. ft.

$$\frac{\text{Total coal}}{\text{Firebox volume}} = \frac{3386.5}{144.25} = 23.476\text{lbs. per cub. ft.} = C.$$

$$\frac{\text{Firebox surface}}{\text{Firebox volume}} = \frac{161.6}{144.25} = 1.12 = K.$$

$$\text{Efficiency of combustion} = \frac{28.5}{25.4 + KC} \quad \frac{28.5}{25.4 + 1.12 \times 23.476}$$

$$= \frac{28.5}{25.4 + 26.29312} = \frac{28.5}{51.69312} = .5532$$

$$\text{Air used per lb. of coal completely burnt} = \frac{6200}{280 + G}$$

$$G = \text{coal fired per sq. ft. grate per hour} = \frac{6200}{280 + 130}$$

$$= 15.122\text{lbs.}$$

Products of combustion per lb. coal fully burnt = air + (coal — ash), say 15.122 + .948 = 16.17lbs.

NOTE.—This means 5.2 per cent. of solid ash remaining on the grate. The average of the L. and Y. coal samples is 5.3 per cent. I have called it 5.2 to get rid of the third decimal.

Total heat per lb. of coal, average of L. and Y. samples = 13664 B.Th.U. per lb.

NOTE.—I have assumed that this calorific value has had the latent heat of the water of combustion at atmospheric pressure deducted from the gross B.Th.U. as we do in our own determinations. This amounts, as a rule, to about 500 B.Th.U. and is not available for heating in a boiler.

Temperature of coal and feed water assumed to be 50° F. (510° F. absolute). Heat, in feed water delivered by the injector being obtained from the heat of the steam evaporated by the boiler, must be debited against the evaporation which is done by taking cold feed.

Total heat required per lb. of steam = 1198 — 18 = 1180 B.Th.U. per lb.

Average specific heat of products of combustion between T and t (T and t absolute F.°) $= .23 + .000015\ (T+t)$.

Heat given to firebox plates by convection when plate surface is at 400° F. or 860° F. abs. $= .0025T^2 - 2.3T + 129$ B.Th.U. per sq. ft. per hour ($T =$ F.° abs.).

Heat given to firebox plates by radiation when plate surface is at 400° F. or 860° F. abs. $= 16T^4/10^{10} - 875$ B.Th.U. per sq. ft. of radiating surface per hour.

For bituminous coal the radiating surface may be taken as being equal to the grate area plus half the firebox heating surface as the flame radiation is somewhere about half as active as that from the solid fuel.

$$\text{Tube efficiency (per one tube or a set)} = \frac{l+56}{35d+l+56}$$

This means efficiency is l when the gases are reduced to steam temperature of 380° F. (840° F. abs.) per 180lbs. steam.

Total heat produced by 1lb. coal fully burnt = 13,664 B.Th.U.

Heat carried away by gases ;—

$$= 16.17 \{ .23 + .000015 \ (T + 510) \} (T - 510 \ \text{B.Th.U.}$$
$$= 16.17 \ (.23T - 117.3 + .000015T^2 - 3.9)$$
$$= 16.17 \ (.000015T^2 + .23T - 121.2)$$
$$= .00024255T^2 + 3.7191T - 1959.804 \ \text{B.Th.U. per lb. coal fully burnt.}$$

Heat given to firebox by convection :—

$$\frac{\text{Firebox surface}}{\text{Coal fully burnt per hour}} = \frac{161.6}{3386.5 \times .5532} = \frac{.086266}{\text{sq. ft. per lb. per hour.}}$$

$$.086266 \ (.0025T^2 - 2.3T + 129)$$
$$= .00021566T^2 - .1984T + 9.328.$$

Heat given to firebox by radiation :—

$$\frac{\text{Radiating surface}}{\text{Coal fully burnt}} = \frac{107.3}{1873.4} = .0578 \ \text{sq. ft. per lb. per hour.}$$

$$.0578 \left(\frac{16T^4}{10^{10}} - 875 \right) = .9248T^4 - 50.575.$$

Heat taken by gases $.00024255T^2 + 3.7191T - 1959.804.$

,, convected $.00021566T^2 - .1984T + 9.328.$

,, radiated $\frac{.9248T^4}{10^{10}} - 50.575.$

$$\frac{.9248T^4}{10^{10}} + .00045821T^2 + 3.5207T - 2001 = 13,664.$$

$$\frac{.9248T^4}{10^{10}} + .00045821T^2 + 3.5207T = 15,665.$$

Temperature of firebox 2,530° F. abs.
460

2,070° F.

Heat carried away by gases :—

$$1552.55 + 9409.32 - 1959.8 = 9,002.$$

Heat absorbed by firebox :—

$$13,664 - 9,002 = 4,662 \ \text{B.Th.U.}$$

Heat retained by gases when cooled to steam temperature:—

$$16.17 \{ .23 + .000015 \ (840 + 510) \} (840 - 510)$$
$$= 16.17 \{ .23 + .000015 \ (1,350) \} \ 330$$
$$= 16.17 \times .25025 \times 330 = 1,335.36 \text{ B.Th.U.}$$

Heat absorbable by tubes = 9,002 — 1,335 = 7,667 B.Th.U.
Inside dia. of tubes, say 11 gauge thick = 2 — .232 = 1.768″
length = 15ft. = 180″

$$\text{Efficiency of tubes} = \frac{56 + 180}{35 \times 1.768 + 56 + 180} = \frac{236}{297.88} = .792265.$$

Heat taken up by tubes = 7,667 × .792265 = 6,074 B.Th.U.
Add heat taken up by firebox = 4,662 ,,

Total heat absorbed per lb. of coal fully burnt = 10,736 B.Th.U.

Total heat absorbable = 13,664 — 1,335 = 12,329.

$$\text{Real heat absorbing efficiency of boiler} = \frac{10,736}{12,329} = 87\%.$$

$$\text{Total evaporation per hour} = \frac{3386.5 \times .5532 \times 10,736}{1,180}$$

= 17,000lbs. per hour.

Maximum sustained i.h.p. of engine at 25lbs. steam per i.h.p. = 680.

When injector is shut off only 1,180 — 330 = 850 B.Th.U. per lb. of steam are required and it will maintain 1,180/850 × 680 = 943 i.h.p.

A compound engine should maintain continuously about

$680 \times \dfrac{25}{20} = 850$ i.h.p., and with the feed shut off about

$$850 \times \frac{1,180}{850} = 1,180 \text{ i.h.p.}$$

But the compound engine would only do this at moderate speeds. At high speeds there is not very much between the compound and simple engines.

The smokebox temperature in the above example is found as under:—

Heat in smokebox gases = 13,664 — 10,736 = 2,928 B.Th.U.

Then $16.17 \{ .23 + .000015 \ (T + 510) \} (T - 510) = 2{,}928.$

$$.23T - 117.3 + .000015T^2 - 3.9 = 181.076$$
$$.000015T^2 + .23T = 302.276$$
$$T^2 + 15{,}333T = 20{,}151{,}733$$
$$(T + 7{,}666)^2 = \underline{20{,}151{,}733} + 58{,}767{,}556$$
$$T + 7{,}666 = \sqrt{78{,}919{,}289}$$
$$T = 8{,}883 - 7{,}666 = 1{,}217^\circ \text{ F. abs.} = 757^\circ \text{ F.}$$

Evaporation at full sustained power $\frac{17{,}000}{3386.5} = 5.02$lbs. steam per lb. coal.

The firebox is responsible for $\frac{4{,}662}{10{,}736} = .435$ per cent. of the total, and the tubes for .565 per cent., that is to say, the firebox evaporates $17{,}000 \times .435 = 7{,}400$lbs. per hour or $\frac{7{,}400}{161.6} = 45.9$lbs. per sq. ft., *i.e.*, 55.8lbs. from and at 212° F., which is practically Cole's value of 56lbs.

With a larger firebox both the volume and surface volume ratios would be improved. Our latest 2-8-0 engines have a volume of 168 cu. ft. and a surface of 168 sq. ft., making a surface volume ratio of 1. The grate is 26 sq. ft. and at 130lbs. per sq. ft. of grate the efficiency of combustion is

$$\frac{28.5}{25.4 + 20.1} = \frac{28.5}{45.5} = .64.$$

The absorbing efficiency will be a little higher than the L. and Y. 0-8-0 as the firebox temperature is higher, but assuming it to be the same, our boiler would have an overall efficiency of $.64 \times .87 = .557$, whereas yours has $.5532 \times .87 = .481$, and our boiler will evaporate 19,500lbs. of saturated steam per hour for the same coal that gives you only 17,000, so that our boiler would give 780 i.h.p. indefinitely sustained to your 680.

The tubes give an efficiency of .792 whereas the total boiler is .87 or 1.1 times the tube efficiency.

This leads to a rough first approximation to finding boiler efficiency which is

$$\frac{28.5}{25.4 + KC} = \frac{56 + l}{35d + 56 + l} \times 1.1 = \text{real efficiency.}$$

The usually quoted efficiency, *i.e.*, $\frac{\text{total heat absorbed}}{\text{total heat in coal}}$ is wrong as in the boiler 1,335 B.Th. U. are unabsorbable.

The usually quoted efficiency of the boiler is $\frac{10{,}736}{13{,}664}$ $= .786$ or practically that of the tubes which gives for the heat taken by the boiler

$$\frac{28.5}{25.4+KC} \times \frac{56+l}{35d+56+l} \times \frac{\text{total heat per lb. coal}}{1{,}180}$$

$$= \text{evaporation at 180lbs.}$$

It gives too low a result for larger fireboxes than the L. and Y. 0-8-0, too high for smaller ones, but the error is not very serious.

$$K = \frac{\text{firebox surface}}{\text{firebox volume}} \qquad C = \frac{\text{total coal fired per hour}}{\text{firebox volume}}$$

l = length of tube in inches $\qquad$ d = inside dia. of tube in inches

The L. and Y. 0-6-0 engine has a firebox volume of 99.5 cub. ft. and a firebox surface of 107.68 sq. ft. and 18.75 sq. ft. of grate. Its efficiency of combustion is therefore

$$\frac{28.5}{25.4 + 1.082 \times \dfrac{18.75 \times 130}{99.5}} = \frac{28.5}{51.8} = .55.$$

Its tube efficiency is

$$\frac{56 + 129.375}{35 \times 1.518 + 56 + 129.375} = \frac{185.375}{238.5} = .77.$$

Its efficiency by the approximate method at a rate of firing of 130lbs. per sq. ft. grate is $.55 \times .77 = .423$ and its evaporation per lb. of coal by the same method is $\frac{13{,}664}{1{,}180} \times .423$ $= 4.9$lbs. steam.

Its total steam production is $4.9 \times 130 \times 18.75$ $= 11{,}950$lbs. and its i.h.p. at 25lbs. per i.h.p. is $\frac{11{,}950}{25}$ $= 478$, *i.e.*, 25.6 i.h.p. per sq. ft. of grate or $\frac{2435.5}{4.78} = 5.05$lbs. of coal per i.h.p.

The larger engine develops 680 i.h.p. per 26.05 sq. ft. of grate or 26.1 i.h.p. per sq. ft. of grate and $\frac{3386.5}{680}$ $= 4.98$lbs. coal per i.h.p. hour. A cubical firebox of the same surface as that of 0-8-0 engine would have linear dimensions of $3\sqrt{144.25} = 5.25$ft. Its grate would be $5^2 5 = 27.5$ sq. ft. and its surface $5 \times 27.5 = 137.5$ sq. ft.

$$\text{Its } \frac{\text{surface}}{\text{volume}} \text{ ratio is } \frac{137.5}{144.25} = .953.$$

Such a firebox, if burning the same quantity of coal per hour as the 0-8-0 in the example given, would have an efficiency of combustion of

$$\frac{28.5}{25.4 + .953 \times 23.476} = \frac{28.5}{47.8} = .597$$

and the overall boiler efficiency by the approximate method is $.597 \times .7922 = .473$. Its evaporation is $\frac{13,664}{1,180} \times .473$ $= 5.48$lbs. water per lb. coal and this is equal to $3386.5 \times 5.48 = 18,600$lbs. per hour $= 742$ i.h.p. The extra h.p. percentage per lb. coal is $\frac{742}{680} - 1 = 9.5$ per cent. as the result of the better *shape* of the firebox.

For identical shapes an increase in linear dimensions has a great effect on increase in efficiency. Assume a 6ft. $\times$ 6ft. $\times$ 6ft. firebox:—Volume = 216 cu. ft.; surface $= 36 \times 5 = 180$ sq. ft.; $S/V = .833$ at 130lbs. per sq. ft. grate;

$$\frac{\text{total coal}}{\text{volume}} = \frac{36 \times 130}{216} = 21.65;$$

efficiency of combustion

$$= \frac{28.5}{25.4 + .833 \times 21.65} = \frac{28.5}{43.45} = .657.$$

With 15ft. tubes 2in. outside dia. $\times$ 11 S.W.G. thick the approximate overall efficiency is $.657 \times .792 = .52$ and the evaporation $= \frac{13,664}{1,180} \times .52 = 6$lbs. water per lb. coal. The i.h.p. $\frac{6 \times 36 \times 130}{1,255} = 1,125$ sustained indefinitely $= 31.4$ i.h.p. per sq. ft. grate and $\frac{36 \times 130}{1,125} = 4.17$lbs. coal per i.h.p. hour.

These evaporative efficiencies may appear low but are quite in accordance with the results obtained from engines tested in various trials at Purdue, Urbana, and elsewhere in America at similar rates of firing. It must be remembered that they do not represent the average efficiency of the engine, but only that when it is making " all out."

The engines can of course develop still higher powers than those quoted, but the coal consumption for such powers will be at a higher rate than 130lbs. per sq. ft. of grate per hour and the evaporations per lb. of coal fired will be lower than those given.

Mr. Billington (L. and Y. Rly., Horwich): In the first place I think we are indebted to Mr. Gass for bringing forward this Paper, which appears to me to be an endeavour to collate all the best information and personal tests in such a way that we can design a locomotive on first principles, form some idea of the work it will do, what it will do it for, and so on. Speaking about testing locomotives, I have had some experience in testing and know how difficult it is to draw conclusions.

There is the question of weather, quality of coal, water, and the human element of the driver and fireman. In the design of the locomotive, it appears to be that the portion at present which requires the greatest attention is the boiler and firebox. There is a saying that the business end of a locomotive is the boiler, but we may go further and say that the business end of the boiler is the firebox. Stationary engineers have shown us how to burn coal economically in the case of the water-tube boiler. Here you have a big combustion space surrounded by hot brickwork. We have not in this country made progress in locomotive boiler design in the proper direction. The combustion chamber now employed largely in America appears to be a step in the right direction, as it is a means of increasing firebox volume which is so essential for burning bituminous coals. The earlier locomotives had deep square fireboxes, and a good deal of their success can be attributed to that fact; but in later years the depth of the firebox has been decreased in order to clear the axle or axles underneath the grate of one or more pairs of coupled wheels. One will have noticed that in some of the later American designs a carrying axle underneath the grate has been resorted to, so as to get a deep firebox. The combination of a deep firebox, a combustion chamber and an efficient brick arch goes a long way to a more perfect firebox. With regard to the starting effort formula recommended by the Author, we have had several long runs with the 4-6-0 passenger engine hauling heavy trains with the dynamometer car attached. Comparing ten good starts, I find that the maximum pull was 10 tons. The pull by the equation is 10.06 tons, which confirms the reasonableness of 82 per cent. of the boiler pressure as the mean effective pressure factor. The resistance of the 4-6-0 engine is assumed to be the same as for the 0-8-0 type.

Mr. E. M. Gass: I am pleased that the contribution has aroused interest and am sure that the excellent discussion which has followed will materially enhance the value of the Paper.

Mr. Smith at the last meeting mentioned that the Paper touches on a very contentious subject, and one is in agreement with him, for no two tests in service will come out alike. You may run the train over the same course, at the same speed, with the locomotive operated in the same manner as regards the combination of cut-offs and regulator openings, and the result will be at variance, due to weather and other conditions. In the course of an hour or so the weather may change, on the first test the locomotive may be faced with a direct head wind, on the next run the wind may be blowing towards the whole side of the train. The data embodied in the Paper is, however, deduced from a large number of experiments carried out in actual service, and ought in a general way to be useful to the designer and to those responsible for computing train loads. The formulæ are only applicable to train loads hauled under normal conditions. To meet abnormal conditions, such as strong gales, etc., the loads must either be reduced or speeds will be correspondingly lower.

With regard to the 14.6lbs. per ton resistance at 11.29 miles per hour, this appears to be high. The resistance of engine, tender, and train by the suggested formulæ works out at 12.8lbs. per ton for a train of 10½-ton wagons hauled by the 0-8-0 engine. For a train of loaded tenders of probably not less than 30 tons weight the resistance would still be further reduced as the load would be concentrated in a fewer number of vehicles.

Respecting the internal resistance figures at starting, *i.e.*, 23lbs. and 16½lbs. for the 0-8-0 and 0-6-0 engines respectively, these efficiencies are from actual tests.

D. L. Cole ("Rating Locomotives") states that the wheel arrangement is the distinguishing feature as regards machinery resistance, and that resistance increases with the number of coupled wheels. For a 0-6-0 engine he appears to suggest 27lbs. and for a 2-8-0 type 31lbs. per ton, probably the latter is also applicable to a 0-8-0 type. These figures include the engine and tender as a vehicle, and the internal resistance of the machinery. The Author has attempted to separate these resistances thus:—

0-8-0 Type.

Total resistance	$95.75 \times 31 =$	2,968lbs.
Resistance as a vehicle	$95.75 \times 18 =$	1,723lbs.
Resistance as a machine	$=$	1,245lbs.
Lbs. per ton $\frac{1,245}{53.75}$ ...	$=$	23lbs.

0-6-0.

Total resistance	$68.25 \times 27 =$	1,842lbs.
Resistance as a vehicle	$68.25 \times 18 =$	1,228lbs.
Resistance as a machine	$=$	614lbs.
Lbs. per ton $\frac{614}{42.15}$ ...	$=$	15lbs.

With reference to Mr. Lang's remarks, the formulæ given are only applicable to single expansion locomotives using saturated steam, and not saturated steam compound locomotives, nor engines using superheated steam of the simple and compound types. Referring to the pressure factor in the tractive effort equation $\frac{D^2 \times S \times MEP}{W}$ for superheated steam, it is usual to assume 65 per cent. when comparing with 80 per cent. in the case of saturated engines. The first superheater engines put in service on the Lancashire and Yorkshire Railway were conversions of the 0-6-0 type referred to in the Paper. The 18in. cylinders were replaced by 20½in., but the boiler only altered to accommodate a superheater. In service the superheater engines are able to start loads 10 per cent. greater than the sister saturated engines. The tractive effort of the former is 21,874lbs., and of the latter 19,886lbs. The mean effective pressure of the superheater engine becomes

$$\frac{61 \times 21,874}{20.5 \times 20.5 \times 26} = 122\text{lbs. or } \frac{122 \times 100}{180} = \text{68 per cent. of the boiler pressure.}$$

A diversity of opinion exists as regards the power of superheated steam and the subject calls for investigation.

Mr. Lang, I am pleased to hear, supports the suggestion that a locomotive test plant ought to be installed in this country, for it is well known (under service conditions) how difficult it is to test locomotives and more difficult still to arrive at conclusions from the conflicting results that the tests reveal. In stationary engine practice many improvements have been made by the establishment of experimental plants, but more might be achieved by the erection of a plant for locomotive experiments. Such an installation would not only be valuable and beneficial for national research purposes, information which could not be questioned, but it would also be of value as a means to the instruction of students.

The first test plant at the Purdue University was begun in the spring of 1891 and ready for the reception of the first locomotive in September of that year.

This plant was destroyed by fire on January 23rd, 1894, and all experimental data lost, and although the disaster entailed a heavy burden of labour and expense, immediate steps were taken for renewal, and four months later a second plant, embodying many improvements, was ready for installation of the second experimental locomotive. Towards the end of 1894 work was begun on the new plant and continued regularly for over two years. During that time more than 50 boiler and engine efficiency tests were carried out under various conditions of load, speed, steam pressure, cut-off, and valve proportions. A study was made of draught action by the exhaust, steam distribution in the cylinders, single and double ported valves, power to move unbalanced and balanced valves, about 10,000 indicator cards were taken, 20,000 miles run, and 25,000 different observations made. The total number of facts was nearly 50,000 available in regard to this one experimental locomotive. I mention these figures to show the accumulation of data necessary to arrive at anything like true facts.

Referring to locomotive development in France. Compounding was first introduced in that country in 1878 when "Mallet" exhibited a small 0-6-0 4-cylinder compound locomotive, the running of which proved that the advantages of double expansion were not solely associated with marine and stationary engines, and since that time great developments in compounding of locomotives have taken place in France, more so than in any other part of the world. Compounding of locomotives has not made much headway in this country, and when first seriously considered about the year 1884, locomotive engineers, although admitting that there was some gain in respect of fuel consumption, still the price of coal at that time was so low, namely, 6s. 3d. per ton, that the resultant economy did not compensate for the extra outlay entailed, due to the compound being more complicated than the simple engine. The price of coal to-day has reached about 30s. per ton, nearly five times what it was in 1884, yet the compound locomotive, so far as this country is concerned, is the exception and not the rule. Now that coal has reached such an enormous price every means ought to be adopted in order to economise even at the cost of complication. French engineers are to be complimented on the manner they have tackled the compound locomotive problem on scientific lines and also for educating the men on the footplate in respect to the correct method of operating this type of engine. France is investigating the question of superheating in combination with compounding. The Paris-Lyons-Mediterranean Railway has built two engines, one with four simple cylinders and one a four-cylinder compound. The two engines are identical in all other respects. The superheated compound in service shows an economy superior to the superheated simple engine of 9 per cent.

Respecting the remark about the shape of the front end; the form and shape of the front surface decidedly influences the resistance offered to the locomotive in passing through air. According to experiments by Borda and later by M. Desdouets, the prow-shaped front offers 40 per cent. less resistance than the flat front at right angles to the direction of movement. At 44 miles per hour the resistance in horse-power would be 46 with the prow-shape and 77 with the flat surface, at 60 miles per hour 116 and 193 respectively. The gain obtained by using the prow would vary in fast trains, according to the speed, from about 30 to 150 h.p.

Mr. Lang referred to the question of valves for reducing back pressure. I should like to draw his attention to the patent automatic ball pressure release valve of Mr. Geo. Hughes which relieves excessive compression by automatically communicating the cylinder and steam chest at the moment that the pressure in the former exceeds that in the latter.

The principle has been applied to both slide and piston valves, and a larger number of the latter are in service.

With regard to lowering the boiler pressure of superheated engines. Superheating is certainly a means of reducing boiler pressure and still retaining equal power with the saturated engine, but from an economical standpoint it is not wise to lower pressure. Some time ago investigations were carried out on the L.Y.Rly. by putting in service a 0-6-0 superheater engine and a sister saturated engine. For the first period the boiler of the former was pressed at 160lbs. and later at 180lbs. per sq. inch, but the saturated engine was worked at 180lbs. throughout. Compared with the saturated engine, the superheater, when working at 160lbs., showed an economy of 12.74 per cent.; the saving, however, increased to 21 per cent. when the pressure was raised to 180lbs. per sq. inch. The results are directly opposed to the theoretical case presented by Dr. Schmidt in the use of high superheat with low pressure. The working pressure of the L. and Y. superheaters is 180lbs. per sq. inch, the same as for non-superheater engines.

Mr. Rowland remarking on the mean effective pressure curves (Fig. 1) states that the reason why a maximum power is maintained up to a higher piston speed in the case of American practice, is due to the fact that American locomotive engineers generally adopt a longer valve travel than we do in this country. That is so, but I do not agree with his statement that long travel is only beneficial on the admission side of the valve. There is little difficulty in getting steam into a cylinder, the trouble is in getting it out. Long travel, in addition to giving a quicker port opening, ensures the port when exhausting being kept more fully open for a longer time than is the case with short travel. This minimises back pressure and results in the area of the diagrams being increased. For high speed running at short cut-offs, long travel is particularly advantageous, and in America as much as $7\frac{1}{2}$ inches is employed, but in this country 5 inches is the general rule.

The curve resistance formula given by Mr. Rowland is probably nearer correct than the simpler equation suggested by the Author, but the latter is sufficiently reliable for arriving at train loads. Mr. Rowland, by his method, estimates a resistance of 11.1lbs. per ton for a 4-wheeled 8ft. wheel base wagon moving round a curve of 5 chains radius. By the equation given in the Paper the resistance is 12.1lbs. per ton.

It is difficult to determine curve resistance for it is effected by so many variable factors. Stone's equation $.1299\sqrt{G^2+B^2}$, G being gauge of line in feet, and B rigid wheel base in feet, the resistance for a 16ft. 4in. wheel base = 2.2lbs. per ton per degree of curvature, for an 8 feet wheel base 1lb. per ton. Under best conditions curve resistance may be as low as .5lbs. per degree and as high as 2lbs., and so as a ready method the Author favours the American Master Mechanics' Association recommendation of 1.4lbs. for locomotives and .7lbs. for wagons per ton per degree of curvature.

With reference to boiler power, I am quite in agreement with Mr. Rowland that for burning bituminous coals a large firebox volume is essential, but it is of the first importance in order to burn enough economically to generate the desired quantity of steam that the grate should be large, and the best ratio between grate and heating surface appears to be about 1 to 65. The Lancashire and Yorkshire Rly. Co. have in service a number of engines possessing a firebox volume of 199 cubic feet and a ratio of grate to heating surface (waterside) 1 to 92. This ratio would certainly have been decreased had it been possible to do so without resorting to an extremely long firebox. Some five or six years ago the Author was privileged to travel on the footplate of one of the large 2-10-0 goods engines on the Belgian State Railway and was greatly impressed by the slow rate of combustion. The engine had a grate surface of 54.8 square feet, a heating surface (including superheater) fireside of 3,249 square feet, ratio of grate to heating surface 1 to 60, firebox volume 288 cubic feet. The following, regarding the performance of the engine, may be of interest :—

Weight of train behind drawbar = 490 tons.
Average speed 17.6 miles per hour.
Rising grade 1 in 62 for 7.9 miles.
Coal consumed per train mile 52.1lbs.
Coal consumed per ton mile .106lbs.

The fireman's work was particularly easy, he only fired on seven occasions during the four hours' running time. The coal (briquettes) put on the grate was 3,850lbs.

Respecting stress in firebox stays, there is no question but that they are subject to a bending stress in addition to the load imposed by steam pressure, and the further the stays are removed from the foundation ring the more pronounced is the bending action. This is one of the advantages of wide water spaces in respect to increased stay flexibility. What I wish, however, to infer is that a one-inch diameter copper stay, supporting an area of 16 sq. inches subjected to a pressure of 180lbs. per sq. inch, equivalent to a load of

1.3 tons, is overloaded. A one-inch diameter stay has at the bottom of the thread a cross sectional area of .6 sq. inch. Assuming, under working conditions, a temperature of 650° F. and an ultimate stress of 10 tons per square inch, then the breaking strength of the stay is 6 tons, giving a factor of safety of 4.6, which is low for good boiler work. Some authorities, however, state that the temperature of the stay head in the hottest part of a locomotive firebox reaches 800° F., and at this temperature the ultimate strength is reduced to $5\frac{1}{2}$ tons per sq. inch, giving a factor of about $2\frac{1}{2}$, but the yield point or elastic limit is the vital test, and this is stated to be 1.8 tons per sq. inch for copper subjected to 800° F. temperature, or a little over 1 ton for a one-inch diameter stay. It would appear from these figures that stays subject to these conditions are working beyond their elastic limit, consequently maintenance will be high.

With reference to Mr. Haigh's remarks respecting air resistance. Some 25 years ago a set of elaborate tests were conducted on the Northern Railway of France in respect to air friction on a locomotive and carriages. The tests were run on a level straight track with a locomotive and ordinary 4-wheeled passenger stock. The results were as follows :—

Speed.	Resistance.	Lbs. per ton.
Miles per hour.	Engine and tender.	Carriages.
20	5.5	2.8
40	9.3	4.0
60	14.5	8.0

The excess which the engine and tender appears to have over the carriages comes mostly from the action of air on the front of the engine, and in a smaller degree from the friction of the motion. These results indicate that resistance is considerable not only due to friction of the sides of the carriages but also to head air resistance, and it would appear possible to reduce this by the pointed front of the locomotive, and also by making the sides of carriages perfectly smooth and as free from projection as possible. The influence of the shape of the front end has already been referred to in answer to Mr. Lang's remarks.

Mr Fletcher asked a question respecting the brick arches of the two engines. The brick arch of the 0-6-0 engine is 2ft. 5in. long, practically half the length of the inside box. It is set at an inclination of 1 in 3, the nearest distance between the arch and the top of grate is 1ft. 9in., and its highest point is level with the centre of the firehole. The length of the brick arch of the 0-8-0 type is 4ft. 5in., approximately 60 per cent. of the length of the inside box.

Its inclination is 1 in $4\frac{3}{4}$, minimum distance between arch and top of grate is 1ft. 8in., highest point 1ft. 3in. above centre of firehole. As regards the best proportions for length and slope opinions differ; what might be satisfactory on one type of engine is often the reverse on another class. It is a matter of experiment. The relation between the air supply, the coal, and the arch is so intimate that they must be considered conjointly.

Regarding Mr. Holcroft's communication. The internal resistance of the two types of engines has been arrived at thus:—

Indicated tractive effort — drawbar pull + resistance of engine and tender as a vehicle.

From a large number of observations we find the minimum internal resistance of a 4-4-0 engine to be approximately 4lbs. per ton.

The train hauled by the two goods engines was composed of a fairly uniform stock consisting of goods wagons of 9ft. 0in. wheel base.

It is quite true that the two engines do haul their classified loads, but not at the speeds specified; for instance, up the grade of 1 in 100 the 0-8-0 engine hauls its load of 580 tons at 10 miles per hour. The lengths of the grades are as follows:—

1 in 100 = 4 miles.
1 in 72 = 5·75 miles.
1 in 65 = ·75 miles.

The tests were carried out with a dynamometer car fitted with instruments for recording drawbar pull, speed, velocity and direction of wind; in conjunction with these observations indicator diagrams were recorded.

Mr. I. E. Mercer in a communication raises a question as to whether it is better from an economical standpoint to use large cylinders and early cut-offs, or smaller cylinders with correspondingly later cut-offs. The former, in my opinion, is preferable for two reasons:—(A) More power is available for starting on a heavy grade. (B) Greater range of expansion which tends to economy. The failure of the L. and Y. Rly. passenger engine was not due to its larger cylinders but to the fact that the boiler was not large enough to meet the demand of these cylinders at high speeds. Our experience goes to show that it is unwise to notch up lower than 17 per cent. cut-off, as below this the indicator diagram develops a loop of negative work without exhaust clearance on valves. For equal volumes the 19-inch cylinder must cut-off at 15 per cent. as compared with 17 per cent. in the case of the 18-inch diameter. This earlier cut-off caused the larger cylinder engines to be sluggish and when operated at a later cut-off the boiler got "out of breath."

Valve gears certainly play an important part and there seems scope for improvement in the direction of quicker opening, longer expansion period, and decreased terminal pressure. Longer duration of the exhaust without resorting to exhaust clearance on valve.

The pitching of stays has already been dealt with.

As regards the blast pipe and smokebox there is no doubt that the volume of the latter can be carried so far that the blast action on the gases is affected; on the other hand, the capacity can be so small as to cause the sharp intermittent blast to draw a considerable quantity of unburnt fuel up the chimney. There is no reliable data as to the exact proportions for each type of engines. Much research and experimenting is wanted. My views in respect to a stationary test have already been expressed. I believe that data obtained from such a plant in conjunction with results in actual service would be far-reaching in solving many of the locomotive problems which unfortunately at present exist.

Mr. Billington remarked that "the business end of the locomotive is the firebox." I believe the success of the engine largely depends on the correct proportion of grate and heating surfaces of this particular part. He also puts in a plea for the combustion chamber now so largely employed in America. The use of the combustion chamber adds materially to firebox heating surface and volume, thus tending to better combustion. It also removes the ends of the tubes from the hottest part of the fire and in addition shortens the tubes which in many boilers are unduly long. It is pleasing to hear that Mr. Billington has investigated a number of starting records of the L. and Y. 4-6-0 passenger engine and finds the M.E.P. factor to be 82 per cent. of the boiler pressure.